Evaporator Modelling Under Conditions of Dry, Wet, and Frosted Finned Surfaces

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ABSTRACT

Three models for finned evaporators are presented. One detailed model, analyzing the coil in geometrically fixed elements, is used to study the local behaviour of the heat transfer fluids and the effect of refrigerant side pressure drop on the heat transfer performance. This model also allows the determination of refrigerant charge. A simplified, and computationally faster model neglecting refrigerant side pressure drop effects on evaporator performance, and any dehumidification in the region of superheated refrigerant is derived for use in evaporator study and design. Finally a parametric model in which coil performance is described by a coil characteristic, and a coil effectiveness parameter is constructed. This model is empirical in nature. It is superior to the first two models in terms of computational speed and is therefore suitable for use in heat pump simulation programs.

The models predict evaporator performance under dry and dehumidifying conditions. For the dehumidifying evaporator, wet and frosted finned surface conditions are considered. The transient frosted evaporator performance is analyzed assuming a quasi-steady state where calculations at one time step can be used for analysis at the next time step. The heat transfer coefficients and frost property correlations used are state of the art in the current literature. A literature survey on the topics of interest to the evaporator modelling is presented.

Computer programs were written (Fortran 77) for the different models, and the models were verified using a digital computer and

results from experimentation on one six row, and one single row evaporator.

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NOMENCLATURE

A_o	total air side surface area	ft^2
A_t	bare tube surface area	ft^2
A_f	fin surface area	ft^2
A_{min}	minimum free flow through area	ft^2
A_{fc}	coil face area	ft^2
A_i	tube inside surface area	ft^2
C	slope of air stream process line	$lb_{m,da}/(lb_{m,wv}\,°F)$
C_{air}	heat capacity of air side	Btu/(hr °F)
C_{ref}	heat capacity of refrigerant side	Btu/(hr °F)
C_{min}	minimum heat capacity fluid	Btu/(hr °F)
C_{max}	maximum heat capacity fluid	Btu/(hr °F)
C_p	specific heat	Btu/(lb_m °F)
$C_{p,a}$	specific heat of air	Btu/(lb_m °F)
$C_{p,da}$	specific heat of dry air	Btu/(lb_m °F)
$C_{p,r}$	specific heat of refrigerant	Btu/(lb_m °F)
CC	coil characteristic	lb_m °F/Btu
CFM	air volumetric flow rate	ft^3/min
d	depth of coil	ft
d_e	diameter of expansion tool	ft
d_{fc}	diameter of fin collar before attached to tube	ft
d_h	hydraulic diameter as defined by equation (3.59.1)	ft
D	mass diffusion coefficient, water vapor in air	ft^2/hr
D_i	inside tube diameter	ft
D_o	outside tube diameter	ft
D_h	hydraulic diameter	ft
D_{eqv}	equivalent fin diameter	ft
f	friction factor	

f_r	refrigerant friction factor	
fpi	fin density; fins per inch	$inch^{-1}$
Ft	frost number (equation 3.30)	
Ft_{ice}	frost number for ice (equation 3.37)	
G_{max}	m_a/A_{min}, air mass flux	$lb_m/(hr\ ft^2)$
G_r	m_r/A_i, refrigerant mass flux	$lb_m/(hr\ ft^2)$
g	gravitational acceleration	32.17 ft/sec^2
h_a	convective heat transfer coefficient on the air side.	Btu/(hr•F ft^2)
h_{dry}	convective heat transfer coefficient on the air side, dry surface	Btu/(hr•F ft^2)
h_{wet}	convective heat transfer coefficient on the air side, wet surface.	Btu/(hr•F ft^2)
h_{fst}	convective heat transfer coefficient on the air side, frosted surface.	Btu/(hr•F ft^2)
h_e	convective heat transfer coefficient based on enthalpy driving potential	Btu/(hr ft^2 (Btu/lb_m))
h_c	contact conductance	Btu/(hr•F ft^2)
$h_{f,a}$	fouling conductance on the air side	Btu/(hr•F ft^2)
$h_{f,r}$	fouling conductance on the refrigerant side	Btu/(hr•F ft^2)
h_r	convective heat transfer coefficient, refrigerant side	Btu/(hr•F ft^2)
h_{sen}	sensible air side convective heat transfer coefficient (= h_a)	Btu/(hr•F ft^2)
h_{lat}	latent air side convective heat transfer coefficient.	Btu/(hr•F ft^2)
h_{tot}	total air side convective heat transfer coefficient.	Btu/(hr•F ft^2)
h_{2p}	two phase convective heat transfer coefficient, refrigerant side	Btu/(hr•F ft^2)

h_{trn}	transition region convective heat transfer coefficient (refrigerant side)	Btu/(hr·F ft^2)
h_{sup}	single phase convective heat transfer coefficient (refrigerant side)	Btu/(hr·F ft^2)
h_m	convective mass transfer coefficient	$lb_{m,da}$/(hr ft^2)
i	enthalpy	Btu/lb_m
i_{a1}	air inlet enthalpy	Btu/lb_m
i_{a2}	air leaving enthalpy	Btu/lb_m
i_{r1}	refrigerant inlet enthalpy	Btu/lb_m
i_{r2}	refrigerant leaving enthalpy	Btu/lb_m
i_w	heat of condensation or desublimation of water	Btu/lb_m
i_{fg}	heat of vaporization	Btu/lb_m
i_{sg}	heat of sublimation	Btu/lb_m
$i_{r,fg}$	heat of vaporization for refrigerant	Btu/lb_m
$i_{r,g}$	saturated refrigerant vapor enthalpy	Btu/lb_m
I	net interference	ft
j	St $Pr^{2/3}$, sensible j-factor	
JFLAG	(see program listings)	
k_1	empirical factor in friction factor expression for a frosted coil	ft^2/lb_m
K_1	empirical factor in frost property correlations, (equation 3.33)	
Kf	$\Delta x \cdot i_{r,fg}/(L \cdot g)$, refrigerant boiling number	
k_f	heat conductivity of fin material	Btu/(hr·F ft)
k_{fst}	heat conductivity of frost layer	Btu/(hr·F ft)
$k_{r,l}$	heat conductivity of saturated liquid refrigerant	Btu/(hr·F ft)
KFLAG	(see three region program listing)	
L	length in direction of refrigerant flow, along one row	ft
L_c	fin collar length in contact with tube	ft

Le	$h_d/(h_m C_{p,da})$, Lewis number	
$\dot{m}_v$	air dehumidification mass flow rate	lb_m/hr
$\dot{m}_r$	refrigerant mass flow rate per circuit	lb_m/hr
m_{frt}	mass of frost on coil	lb_m
n	number of tube rows	
NTU	UA/C_{min}, number of transfer units	
P	absolute pressure	lb_f/in^2
P_a	air absolute pressure	lb_f/in^2
P_{ev}	evaporation pressure ($\approx P_{sat}$)	lb_f/in^2
P_{sat}	refrigerant saturation pressure	lb_f/in^2
P_v	partial water vapor pressure in air	lb_f/in^2
$P_{vs,m}$	saturated partial water vapor pressure at mean surface temperature on coil.	lb_f/in^2
Pr	$\mu C_p/k$, Prandtl number	
q	heat transfer rate	Btu/hr
q_{avl}	available heat transfer rate in a region of the coil	Btu/hr
q_{lat}	heat transfer due to phase change	Btu/hr
q_{sen}	heat transfer due to temperature change	Btu/hr
q_{tot}	total heat transfer, sensible and latent	Btu/hr
r	C_{min}/C_{max}, fluid heat capacity ratio	
R	ratio of equivalent fin diameter to outside tube diameter	
R_v	ideal gas constant for water vapor	$lb_f ft/(lb_m {}^\circ F)$
Re	$G D/\mu$, Reynolds number	
Re_D	Reynolds number based on the tube outside diameter (equation 3.6.5)	
Re_L	Reynolds number based on longitudinal tube spacing (equation 3.7.1)	
Re_o	Reynolds number based upon free flow through area between fins (equation 3.34)	
Re_r	refrigerant flow Reynolds number	

Res_a	air side thermal resistance	(hr•F ft^2)/Btu
Res_{fst}	frost layer thermal resistance	(hr•F ft^2)/Btu
Res_c	tube fin contact resistance	(hr•F ft^2)/Btu
$Res_{f,a}$	air side fouling thermal resistance	(hr•F ft^2)/Btu
$Res_{f,r}$	refrigerant side fouling resistance	(hr•F ft^2)/Btu
Res_r	refrigerant side thermal resistance	(hr•F ft^2)/Btu
$Res_{a,s,m}$	air side thermal resistance not accounting for fin effectiveness	(hr•F ft^2)/Btu
St	$h/(G\ C_p)$, Stanton number	
SUMR	summation of thermal resistansce terms	(hr•F ft^2)/Btu
s_o	fin separation distance	ft
s	distance between frost surfaces on adjacent fins	ft
t	time	hr
Ti	time number (equation 3.29)	
T_a	air temperature	•F
T_{a1}	entering air temperature	•F
T_{a2}	leaving air temperature	•F
T_{adp}	apparatus dew point temperature	•F
$T_{a,r}$	$T_a - T_r$, air to refrigerant sensible heat transfer driving force.	•F
T_{av}	average air temperature	•F
$T_{c,m}$	mean coil cooling temperature	•F
T_{ev}	evaporation temperature (= T_{sat})	•F
T_{fb}	fin base temperature	•F
$T_{f,m}$	fin mean temperature	•F
$T_{s,fb}$	condensate or frost surface temperature at the fin base	•F
$T_{s,m}$	mean coil surface temperature	•F
T_{sat}	refrigerant saturation temperature	•F
T_r	refrigerant temperature	•F
T_{r1}	entering refrigerant temperature	•F

T_{r2}	leaving refrigerant temperature	°F
UA	overall heat transfer coefficient	Btu/(hr·F)
v	specific volume	ft^3/lb_m
v_f	specific volume of saturated liquid refrigerant	ft^3/lb_m
v_{fg}	specific volume change for evaporation of refrigerant	ft^3/lb_m
V_{fc}	coil face velocity	ft/min
V_{loc}	local air velocity at minimum flow area as defined by equation (3.35)	ft/min
Vm	heat resistance number (equation 3.33)	
w_a	specific humidity of air	$lb_{m,wv}/lb_{m,da}$
w_{a1}	specific humidity of inlet air	$lb_{m,wv}/lb_{m,da}$
w_{a2}	specific humidity of leaving air	$lb_{m,wv}/lb_{m,da}$
w_{av}	average specific humidity of air	$lb_{m,wv}/lb_{m,da}$
$w_{s,m}$	saturation specific humidity at mean surface temperature on coil.	$lb_{m,wv}/lb_{m,da}$
$w_{s,f,m}$	saturation specific humidity at mean mean fin surface temperature	$lb_{m,wv}/lb_{m,da}$
$w_{s,fb}$	saturation specific humidity at surface temperature at fine base.	$lb_{m,wv}/lb_{m,da}$
x	refrigerant vapor mass quality	
x_{in}	inlet quality	
x_{out}	outlet quality	
x_o	refrigerant quality at inlet to transition region	
X_L	longitudinal tube spacing	ft.
X_T	latitudinal tube spacing	ft
y	fin thickness	ft
z	fin height	ft

Greek symbols

β	mass transfer coefficient	ft/hr
ϵ	coil heat transfer effectiveness	
ϵ_{i}	coil enthalpy effectiveness	
ϵ_{T}	coil temperature effectiveness	
ΔL	length of an element	ft
ΔP_{air}	air side pressure drop	lb_f/in^2
ΔP_{int}	initial, nonfrosted surface air side pressure drop	lb_f/in^2
Δt	time step	hr
ΔT_c	temperature drop across fin tube bond	°F
ΔT_{lm}	log mean temperature difference	°F
ΔT_{sup}	refigerant degree of superheat	°F
Δx	quality change	
δ_{fst}	frost layer thickness	ft
δ_w	tube wall thickness	ft
σ	$A_{min}/A_{f/c}$, ratio of minimum flow through area to face area.	
ϕ	temperature fin efficiency	
ϕ_o	temperature fin effectiveness	
$\hat{\phi}$	specific humidity fin efficiency	
ρ_{fst}	density of frost layer	lb_m/m^3
ρ_{ice}	density of ice	lb_m/m^3
ρ_{va}	water vapor density	lb_m/m^3
$\rho_{vs,m}$	saturated water vapor density at mean coil surface temperature	lb_m/m^3
ρ_r	refrigerant density	lb_m/m^3
μ	dynamic viscosity	$lb_m/(ft\ hr)$
μ_g	dynamic viscosity of saturated vapor	$lb_m/(ft\ hr)$
μ_f	dynamic viscosity of saturated liquid	$lb_m/(ft\ hr)$
μ_r	dynamic viscosity of the refrigerant	$lb_m/(ft\ hr)$

$\mu_{r,l}$	dynamic viscosity of saturated liquid refrigerant	$lb_m/(ft\ hr)$
φ	air relative humidity	

Subscripts

a	air
avg	average
dry	dry surface
da	dry air
el	element
est	estimated
f	fin
fr	friction
fst	frosted surface
int	initial
lat	latent
r	refrigerant
s	surface
sen	sensible
spc	specified
sup	superheated
tot	total
trn	transition
wet	wet surface
ref	reference state
2p	two phase

CHAPTER 1

INTRODUCTION

As energy prices have risen, increasing interest in heat pump applications has been evident. The most common heat pump system has been the air source unit. The reason for this is the lower system cost (compared to other systems using alternate sources), and the fact that, with relative ease, the air source unit can be reversed, and used as an air conditioner during the summer months. To predict the performance of a heat pump, one is forced either to bench tests, or to mathematical modelling. In a heat pump model each component is described by a set of equations. The accuracy of the heat pump model obviously depends on the accuracy of the component modelling. This thesis investigates heat pump evaporators. Suitable equations, located in the literature, describing the various physical processes are used to construct evaporator models. Evaporator coil performance can thus be investigated and optimized by numerical simulation which reduces, or eliminates the need for expensive testing procedures.

Three evaporator models have been developed. Two of these models are semi-analytical, and the third is purely empirical. A semi-analytical model consists of governing equations whose form are derived from considerations of scientific laws; however, some empiricism is required at certain points due to incomplete theory, or very complex governing equations. For the heat exchanger analysis empiricism is required in the prediction of heat transfer coefficients and friction factors in refrigerant tubes and across coils. An empirical model is based purely on experimentally, or by other means, generated data.

All three models are complete in the sense that they handle all possible coil surface conditions. These are: dry surface, wet surface, and frosted surface. Water vapor is transported against a pressure gradient. In other words it flows from a region of high vapor pressure to one at lower vapor pressure. Hence the air flowing over the coil will be dehumidified when the temperature of the coil surface is below the temperature corresponding to saturation for the given partial water vapor pressure. Or expressed in another way, whenever the temperature of the coil falls below the dew point of the air stream. If the temperature of the coil is below the dew point and the freezing point of water; the desublimation process will take place and frost will form on the coil surface. When frost is forming on the coil surface a time simulation is required. This case is considerably more complex than the dry, and wet surface conditions. The reason being that the heat transfer and air flow depends on the coil frost build up, and the properties of the frost on the coil in turn depends on the temperature and heat transfer history of the coil.

The parameters specified for each model, are: geometry (e.g. tube rows ,fin density, tube spacing, etc.), inlet air condition, inlet refrigerant quality, evaporation temperature, and refrigerant outlet condition (saturated or superheated). The model then predicts the refrigerant mass flow rate, heat transfer rate, air dehumidification rate, and leaving air condition. Alternatively if the refrigerant mass flow rate is a specified parameter then the refrigerant condition at evaporator outlet is determined. For a frosted coil, frost thickness and frost accumulation is also determined. The air inlet condition and air mass flow rate are determined by the outside weather conditions, and the fan characteristic, respectively. The air condition is specified by, the dry bulb temperature, and the relative humidity. The refrigerant inlet condition is determined by the condenser pressure, and condenser outlet condition (saturated or subcooled) which are functions of the heat pump component characteristics.

The evaporator temperature is a function of heat pump component characteristics. The refrigerant condition at the outlet of the evaporator depends on the type of throttling device used. The commonly used throttling devices are thermostatic expansion valves, and capillary tubes. Limited refrigerant charge is sometimes used, as well. In this work a thermostatic expansion valve was used. The heat pump system used in the tests was equiped with an interchanger. If the expansion valve sensor is placed at the outlet position of the evaporator a superheated outlet condition results. If the sensor is placed to sense the condition leaving the interchanger, the evaporator leaving condition was found to be either saturated (refrigerant quality less than or equal to one) or superheated.

These models were derived to simulate plate finned tube evaporator coils used in heat pump systems. One very detailed finite element model was constructed. This model allows the user to study the behaviour of the heat exchanging fluids at a number of predetermined positions along a row of the coil. A second, simplified, model was then written; the three region model. This model divides the coil into three regions, the extent of each region depending on the state of the refrigerant as it passes through the coil. This is in contrast to the finite element model which has geometrically fixed elements. The three regions are: saturated, transition, and superheated refrigerant region. The three region model contains some assumptions not present in the finite element model; however these assumptions proved to be very reasonable, since the predictions compared well with those of the computationally longer finite element model. Some features, however, are unique to the finite element model. Using the detailed finite element model the mass of refrigerant in the coil can be determined. The finite element model also accounts for the refrigerant side pressure drop, which was found to be very small.

for all mass fluxes of interest. One of the objectives of this work was to derive a short and computationally efficient evaporator model. The finite element and three region models were the necessary tools for the analysis required in constructing a short model. This model has been termed the parametric model. The required data for the parametric model was obtained from the three region model after that model had been verified against the finite element model and experimental results from tests on evaporator coils. The main requirement of the parametric model is computational speed since it is intended to be part of a larger heat pump simulation program which requires a time consuming iterative procedure. The heat pump simulation program is run on a personal computer, demanding both a short and computationally fast evaporator model.

The finite element, and the three region models can be used as tools for obtaining short evaporator models for use in larger simulation programs, and for simulations in an evaporator design process. The models allow the designer to optimize design with respect to: fin spacing and coil frost up, (important for defrosting criteria), air flow rate, number of tube rows, refrigerant degree of superheat or saturation,and any other geometric coil parameter. The advantage of these models compared to most others is that they can handle any environmental condition. For air source heat pumps operating in cold and wet winter climates the problem of frosted coil surfaces is common. Hence these models should be of interest to any designer or researcher in the field of heat pump applications at subfreezing temperatures.

Due to the many significant influence parameters, and the very complicated flow situations, many of the equations describing the thermal and mechanical processes were obtained after extensive experimentation by many investigators in the field. Through the years some areas have been explored more succesfully than others. The literature has been extensively researched to yield suitable equations and theories accurately predicting the behaviour of an evaporator coil, in particular the coils tested in model validation. Because of differences in test conditions and experimental procedure no single universal formula for predicting any of the important physical variables was singled out. Although, some appear to be of more general applicability than others. The first part of this report contains a literature survey. The work of various investigators are discussed briefly, with special emphasis on results applicable to the evaporator models developed. Literature of interest to evaporator modelling, but not used directly in this work, is given in the Bibliography, whereas all referenced material can be found under References. The theories of heat transfer (single and two phase), of presure drop across coils and inside tubes, and of frost property determination are outlined in detail in the following chapter, with the correlations of relevance to the evaporator models explained in detail. When the theoretical foundation has been laid, the three evaporator models, with assumptions and limitations, are discussed. The models are then verified against experimental results. The algorithms for the computer program implementation of the three region and finite element models are given in appendix 2.

The refrigerant and psychrometric subroutines available at this time at Concordia University are in the conventional unit system. Since the majority of the North American literature on this subject upto date is also in the conventional unit system, it was decided to use this system of units in this thesis. Exceptions are made for work done entirely in the SI unit system, particularly where correlations are presented. The use of SI units will be pointed out as they appear.

CHAPTER 2

LITERATURE SURVEY

2.1 Air Side Heat Transfer and Pressure Drop

McQuiston [1981] discussed the air side heat transfer phenomena. The effect of air dehumidification, tube rows and fin spacing was considered. The Colburn analogy, relating the total j-factor ($h_e(Sc^{2/3})/G_{max}$) and the sensible j-factor (St $Pr^{2/3}$), was also investigated. The results, however, show that the analogy is poor. McQuiston noted that for the dehumidifying coil, the water on the surface has a roughness effect which is important when the Reynolds number is high and the boundary layer is thin, but has little effect when the Reynolds number is low and the boundary layer is thick. Droplets on the surface was found to have more effect than a film. The condensate layer affects the heat transfer, friction pressure losses, and mass transfer. Correlations for the effect on these parameters were presented.

McQuiston [1981] plotted the heat transfer coefficient data from his previous investigations, and those of Rich [1973,1975]. A correlation for a four row coil is given in graphical form. This correlation was found to best fit the performance of the coils modelled in this thesis and more details are given in section 3.2. McQuiston's analysis of the investigation of Rich [1975], allows a correction for coils with other than four rows to be made.

Briggs and Young [1962] presented correlations for heat transfer coefficients and friction factors on triangular pitch banks of finned tubes. Their correlations involve a large number of tests and are based on tubes having a wide variations in fin height, fin thickness, fin spacing and root diameters. Briggs and Young noted from data taken from banks with two, four, and six tube rows that within experimental error the isothermal pressure drop per row was independent of the number of rows in the direction of flow. Although the results of Briggs et al. might not be directly applicable to the plate finned tube, many of the observed trends and conclusions will apply.

Elmahdy and Briggs [1979] developed a correlation for dry surface finned tube heat transfer coefficient. Experimental heat transfer data of twenty different heat exchangers were used in deriving the correlation. The heat exchangers used were of the type, tube banks of staggered circular tubes, four rows or more, with circular or continous flat plate fins. Elmahdy's correlation involves the ratios of fin thickness to fin height, hydraulic diameter to fin thickness, and fin spacing to fin height. Elmahdy noted that other less influential dimensionless groups may be identified using more data in the regression analysis.

Turaga, Lin, and Fazio [1988] presented correlations for heat transfer and pressure drop factors for direct expansion air cooling and dehumidifying coils. Ten different coils were used in the experimentation. The coils were of the staggered row, flat plate fin type. The air side correlations in Turaga et al's. work were obtained using an evaporating refrigerant on the coolant side. This is in contrast to the work of Elmahdy and Briggs [1979] and McQuiston [1981] which both used chilled water. Correlations for the dry and wet heat transfer factors are given. The dry heat transfer factor is simply the j-factor and the wet heat transfer factor is equivalent to the total j-factor used by McQuiston [1981]. Friction factor correlations for both the wet and dry surfaces are also given. Turaga et al. illustrate in tabular form the mean deviation of the computed data as compared to experimental data. Turaga et al's. results have a lower deviation for both friction and heat transfer factors than those of Elmahdy and McQuiston.

Webb [1980] surveyed the literature on the air side aspect of heat exchanger design. Webb discussed correlations for heat transfer, and pressure drops, and the effects of fin spacing and tube rows. Recent developments on the enhanced heat transfer due to surface effects is discussed. Local air side heat transfer coefficients in a coil is also considered. Both circular finned tubes and plate fin designs are considered.

Rich [1975] made heat transfer measurements on 6 plate fin and tube heat exchangers, with the flow depth varying from 1 to 6 rows. Air velocities where in the range 200 to 2000 fpm. Rich concluded that at high Reynolds numbers, the heat transfer coefficient for the downstream rows is higher than for the upstream rows. Also the average heat transfer coefficient for deep coils is higher than for shallow coils. Conversely, at low Reynolds numbers, in the range of interest for heat pump coil applications, the heat transfer coefficient for deep coils is substantially lower than for shallow coils. Rich [1975] did not investigate the effect of tube rows on friction. McQuiston [1981] noted that the literature is vague on this point. McQuiston noted, however, that the friction factor does not seem to be influenced very much by the number of tube rows, and stated the assumption that since an expansion and a contraction occur for each row, the friction factor is the same for each row. This assumption is confirmed by Briggs and Young's [1962] experiments as noted above.

In earlier work Rich [1973], investigated the effect of the fin spacing on heat transfer coefficient and fin friction. He noted that with few fins the friction factor varies only slightly with Reynolds number, but as fins are added there is a progressive increase in the variation with Reynolds number. It was determined that form drag accounts for a significant portion of the total pressure drop. Rich stated that the form drag does not appreciably depend on the number of fins present, and hence the total pressure drop does not increase in proportion to total surface area. Rich determined that the heat transfer coefficient is essentially independent of the fin spacing within the range from 3 to 21 fins per inch at a given air mass flux, other parameters held constant. He broke up the pressure drop into two components; one due to the tubes, and the other due to the fins. The friction factor for the fins was found to be independent of fin spacing for pitches between 3 and 14 fins per inch at a given air mass flux other parameters being equal. Rich gave correlations for the fin friction factor and the sensible j-factor for the four row coils investigated, in terms of the Reynolds number based on the length of the core in air flow direction. He noted that the friction factor for the tube component may be approximated by the pressure drop in an equivalent prime tube bundle.

Eckels and Rabas [1987], discussed the effect of air dehumidification on the sensible heat transfer coefficient. Many authors have discussed the effect of the added surface roughness, due to the water film, on the heat transfer coefficient. Eckels and Rabas noted that no unique dependence of heat transfer augmentation on film roughness has been established. Eckels and Rabas studied the effect of the transverse velocity of the condensing phase occuring for a dehumidifying coil, on the friction, mass and heat transfer. They noted that the effect is an increase in all these parameters, and pointed out that the increases are analogous to those observed in boundary layer control by suction at the wall. They concluded that the effect is significant, and that progress toward correlation has been made; however, final correlation has not been accomplished. They claimed that all of the sensible heat and mass transfer augmentation and a high fraction of the increased pressure drop is attributable to the transverse velocity of the condensing phase.

A review of the literature on the heat transfer and pressure drop performance of a frosted coil was presented by Kondepudi

[1987]. Many of the early investigators studied the effect of the frost on the overall heat transfer coefficient, U_o. Both Stoecker [1960], and Hosoda and Uzuhashi [1967], noted that there is a small initial increase in the overall heat transfer coefficient. As the frost continues to grow the overall heat transfer coefficient keeps decreasing. Stoecker [1960] noted that the value of U_o jumps by about 20% with the first appearance of frost. Stoecker attributed this to the rough frost surface and the increased heat transfer area. Stoecker did not comment on the surface heat transfer coefficient, but it is clear that it must be enhanced since the overall heat transfer coefficient value initially was. Hosoda et al. [1967] noted that the frost surface heat transfer coefficient can be considerably larger than the dry surface value. They noted from their tests that when there is frost on the surface, the heat transfer coefficient will be approximately twice the value of the dry surface. Hosoda et al. also noted that the pressure drop from the frosted surface is approximately twice that obtained from calculating the pressure drop assuming a smooth dry surface, Hosoda et al. presented correlations for both pressure drop and heat transfer coefficient. As noted by Kondepudi [1987] Hosoda et al's. correlation for overall heat transfer coefficient does not involve any relationship with surface temperatures, air humidity, and air temperature. It is therefore believed that Hosoda et al's. results apply only to the equipment and the conditions under which the tests where carried out. Sanders [1974] noted from experimental observations that the heat transfer coefficient is affected by the roughness of the frosted surface, especially for small frost layer thicknesses. Sanders concluded from his survey of data in the literature that the heat transfer coefficient on a frost surface is higher than the heat transfer coefficient on the comparable smooth surface. Sanders [1974] noted that the measured heat and mass transfer coefficients for different circumstances are approximately 60% higher than the smooth wall values. He noted that the surface characteristics will change as the frost builds up. The heat transfer coefficient will, therefore change accordingly. Sander's observed that after the initial increase the heat transfer coefficient attains a more or less stable value.

Gates, Sepsy, and Huffman [1967] studied the effect of frost formation on heat transfer and pressure drop for a number of different coil geometries. Also they noted that the initial frost build up is beneficial since the rough air side increases the air side heat transfer coefficient. They noted, however, as the frost builds up the pressure drop becomes significant and the operation of the coil eventually becomes uneconomical. Gates et al. presented curves for the adverse effect of frost on the heat transfer coefficient and the friction factor for a number of different coil configurations tested. A nondimensionalized representation of the heat transfer coefficient, which includes the thermal conductance of the frost layer, and the friction factor against dimensionless time and dimensionless humidity are given.

Among the most recent work on frosted finned tubes is that of Malhammar [1986]. The main objective of his work is frost property correlations. Malhammar also used his results to determine pressure drops and heat transfer coefficients for frosted coils. Malhammar noted that due to the uncertainty in many of the measurements the correlations are bound to be rather crude. He assumed an unaffected surface heat transfer coefficient for the frosted coil, and a friction factor related to the frost accumulation per unit coil surface area. Malhammar's work appeared of most general applicability and is discussed in some detail in Chapter 3.

2.2 Refrigerant Side Heat Transfer, Pressure Drop and Effects of Oil.

Pierre [1955] made extensive tests to determine the heat transfer coefficient of evaporating refrigerants. His tests were mainly carried out with Refrigerant-12, but results of other investigators for Refrigerant-11 were also used. Four evaporators of type double pipe heat exchangers were used. In tests of evaporator temperatures below 32 °F an antifreeze solution was used in the outer pipe. The mass and heat fluxes were held within region of interest for cooling and heating applications. Heat fluxes in the range 370 to 7400 Btu/(hr ft^2) were used and, the mass flux was in the range 12,000 to 250,000 lb_m/(hr ft^2). The two phase heat transfer coefficients obtained were in the range of 20 to 450 Btu/(hr°F ft^2). Among the correlations available in the literature, Pierre's correlation appeared most suitable for the modelling of the coils investigated in this work. The terms in Pierre's correlation are discussed in some detail in section 3.5. Andersson, Rich, and Geary [1967] noted that Pierre's correlation appeared to be of the widest scope for the evaporation of refrigerants out of 7 different correlations tested. Andersson et al.'s test results show excellent agreement with the values predicted by Pierre's correlation at low mass velocities. Andersson et al.'s evaporator was also of the doubble pipe heat exchanger type.

Nussbaum discussed evaporation of Refrigerant-12 in an article translated from the German by Polke [1963]. Nussbaum discussed the various terms in the correlation by Pierre. He also discussed the flow mechanism in evaporator tubes. Nussbaum noted that findings by previous investigators indicate that a laminar flow pattern with little wetting occurs at low rates of heat flux density. Definite separation of liquid and gas at low rates of heat flux density (which is often the case in air cooling units) had been noted. It was suggested that non wetted surfaces participate in the heat transfer process only to a small extent so that the heat transfer coefficient in such areas are very low.

Shah [1976] developed a chart correlation, a graphical method of solution. Shah compares the correlation with 800 data points from 18 independent experimental studies. Shah states that the chart appears to be applicable to saturated boiling, inside pipes, of all Newtonian fluids (except metallic fluids) over the entire practical range. He noted that the data for boiling water, taken from five different sources , are all very well correlated with very few points varying beyond ± 30% of meassured values. The larger deviations were observed for refrigerant evaporator data. Shah explained this by noting that refrigerant evaporators frequently contain impurities, in particular oil. He also noted that there is considerable uncertainty regarding the properties of refrigerants. The value given in different sources can differ by significant amounts. Shah commented that the Pierre correlation appear satisfactory for Refrigerant-12 and Refrigerant-22 evaporators in the range of parameters covered in Pierre's experiments; however, outside this range its applicability is doubtful, as is evident from results of previous investigators. Regarding the effect of oil in the refrigerant Shah stated that according to Pierre if heat transfer coefficients were calculated on the basis of saturation temperature of pure refrigerant, up to 18% of oil circulating with R-12 had no perceptible effect. Shah stressed, however that there is not general agreement on the effects of oil and other litterature should be consulted.

Schlager, Pate, and Bergles [1988] studied the evaporation of refrigerant oil mixtures in a smooth tube and a micro fin tube. For both smooth, and micro fin tubes small quantities of oil were found to enhance the evaporation heat transfer coefficient compared to pure refrigerant evaporation. Schlager et al. showed the effect on evaporative heat transfer coefficient in a smooth tube for three different oil concentrations: 1.2%, 2.5%, and 4.9%. The heat transfer coefficient showed an increase for all oil concentrations. The mass fluxes for Schlager et al.'s. tests were on the high side (70,000 $lb_m/hr\ ft^2$ and up), if compared to the range of operation for the coils used in the verification of the models developed in this work (22,000 - 70,000 $lb_m/hr\ ft^2$). From the figures presented by Schlager et al. at 73,500 $lb_m/hr\ ft^2$ the 2.5% oil mixture heat transfer coefficient was enhanced by a factor of 1.35 as compared to pure refrigerant. At oil concentrations of 1.2 and 4.9% the enhancement was approximately 1.15. A peak in the heat transfer coefficient at about 2.5% was observed. At higher concentrations the enhancement decreased. Schlager et al. noted that the enhancement of evaporation with the addition of small amounts of oil, as well as occurence of a maximum enhancement at approximately 1-3% agrees with the conclusions of several previous investigators.

Chaddock [1986] investigated the local evaporative heat transfer coefficient of halacarbons Refrigerant-12, Refrigerant-22, and Refrigerant-502, and for ammonia Refrigerant-117 with and without small fractions of oil present. Chaddock's tests were done at mass flux values considerably higher than those observed in the verification of this thesis; however Chaddock's results agree, according to Chaddock, with the results of Worsoe-Schmidt whose tests were run at mass fluxes of interest to this work. Chaddock observed that the presence of 1% of mineral oil in the refrigerant results in an increase in the local evaporative heat transfer coefficient, compared to oil free refrigerant. The enhancement was observed in the vapor fraction region 0.05 - 0.70, and sometimes to complete evaporation. The coefficients were found to be 20-30% above those of pure refrigerant. Chaddock then increased the oil fraction and noted that for concentrations from 3 - 6% (by weight), the enhancement effect is even greater in the beginning portion of the evaporator, upto vapor fractions of approximately 0.5 . A much higher resistance to heat transfer was, however, observed in the latter part of the evaporator. The effect of oil in the refrigerant on the local evaporative heat transfer coefficient was presented graphically by Caddock [1986]. Chaddock also included the results of Worsoe-Schmidt. Although Chaddock's results were obtainined with Refrigerant-22, as opposed to Worsoe Schmidt who used Refrigerant-12, at mass fluxes 10-20 times higher than those of Worsoe-Schmidt the trends are very similar.

Hughes, McMullan, and Morgan [1982] showed that calculations made using published refrigerant data can be very misleading since these tables do not account for refrigerant oil solubility effects. Hughes et al. noted that the presence of oil in a vapor compression refrigeration or heat pump system reduces its capacity. This is explained by the fact that refrigerant remains dissolved in the oil at the evaporator outlet, and is therefore unavailable for evaporation, which results in reduced capacity. Hughes et al. claim that the common usage of pure refrigerant tables appears to be an insufficiently accurate approximation for oil concentrations greater than 1%. The effects of the oil, according to Hughes et al., are that boiling begins in the mixture at a lower pressure, and a higher enthalpy; there is no distinctive boiling point; the saturated vapor line disappears; and a significant level of superheat appears at much lower mixture enthalpies. Hughes et al. illustrated the effect of using the same meassurements for pressure, temperature and mass flow rate to determine the system performance using curves for pure refrigerant enthalpy, and a curve for an 8% oil mixture. The effects, according to Hughes et al., are reduced evaporator capacity and increased compressor work, which is evident from the graphs presented by Hughes et al.

Pierre [1964] carried out a series of experiments to investigate the pressure drop of Refigerants 12 and 22. Pierre gave two correlations, one for oil free refrigerant and one for refrigerant containing oil. Oil free refrigerant is defined as tests where an oil separator was in use and the oil content was less than 0.5 vol% of liquid mixtures. When oil separators were not in use the oil content was in the range 6-12 vol%. Pierre observed that the oil had the effect of almost doubbling the friction factor. Anderson, Rich, and Geary [1966] compared two refrigerant friction pressure drop correlations: Pierre's correlation and a correlation of Martinelli and Nelson. They considered the correlation given by Pierre for oil free refrigerant, and noted that the Martinelli-Nelson correlation overestimates the pressure drops observed in Anderson et al's. experiments, and that Pierre's correlation underestimates the pressure drop. The selection of the correlation for oil free refrigerant appears correct since Andersson et al's system used an oil separator. Neither correlation is recommended over the other by Andersson et al.. Turaga, Lin, and Fazio [1988] presented a correlation for refrigerant pressure drop. Turaga et al. used the same dimensionless parameters in their correlation as Pierre [1964]. Turaga et al. noted that the agreement between experimental and computed data for the friction factor is ±20% for 90% of the data. They also noted that the results from Pierre's correlation for oil free refrigerant are 30 to 40% lower than those calculated using Turaga et al.'s formula. Turaga et al's tests were made with refrigerant containing two percent oil, and according to Pierre's findings one would therefore expect the predictions using the correlation for oil free refrigerant to fall short of the results for friction factor for two percent oil bearing refrigerant.

Wallis [1969] discusses one dimensional two phase flow theory. It was noted that if the two phase fluid flow is homogeneous the mixture can be treated as a pseudo fluid that obeys the usual equations of single component flow. This is the method used in this work and is further discussed in section 3.9.

2.3 Contact Resistance

In a series of papers, Scheffield, Sauer, and Wood [1987], Scheffield, Abu-Ebid, and Sauer [1985], Wood, Scheffield, and Sauer [1987 "Effect of"], and Wood, Scheffield, and Sauer [1987 "Generalized "], finned tube contact conductance was investigated. The contact conductance is primarily an issue for coils on which the fin tube bond is achieved through mechanical expansion. The fins on the coils investigated in this work are bonded by mechanical expansion. Sheffield et al. [1985] conducted tests on eight mechanically expanded finned tube heat exchangers with different geometries in a dry vacuum environment. A correlation for the dry contact conductance was obtained, (see section 3.6). Wood et al. [1987 "Generalized"] noted that many air side thermal conductance correlations include the effect of contact conductance by default, but to be able to predict coil performance over a wide range of design parameters the air side thermal conductance, and contact conductance should be seperated. Scheffield et al. [1987] noted that neglecting the contact resistance results in an inflated air side resistance of 10-30%. Wood et al. [1987 "Effect of"] conducted tests on 31 coils. All coils except one were of the type mechanically expanded, interference fit, plate finned coils with fin collars. Wood et al. [1987 "Effects of"] found that contact conductance increased as the fin thickness increased. Contact conductance also increased as fin density increased. Contact conductance was found to decrease as tube diameter was increased. Wood et al. concluded from their study that contact resistance is of more importance to the manufacturing engineer rather than the heat transfer design engineer. They noted that from a thermal point of view contact resistance is not significant and determined accurately enough to accommodate the determination of the contact resistance independently from the air side resistance. Wood et al. noted, however, that although contact resistance not significantly affects the heat transfer, it is an indication of product quality.

2.4 Properties of Frost

Hayashi et al. [1977] classified the frost growth into three periods: the crystal growth period, the frost layer growth period, and the frost layer full growth period. During the crystal growth period a high density layer is initially formed. After a short period of time a decreasing frost density is observed. At the transition point between the crystal growth period and the frost layer growth period the frost density is at a minimum. In the frost layer growth period the frost density increases as the frost expands more in the horisontal direction rather than the vertical. Hayashi et al. defined the frost layer full growth period as a period when the frost does not change its shape, especially until the frost surface comes to 32°F. The frost then melts causing a drastic increase in frost density as the melted water is soaked up in the frost layer and freezes deeper in the frost. The densification process described above was also observed by Malhammar [1986] and Sanders [1974]. The frost structures described by Hayashi et al. were in certain cases also observed by Malhammar [1986]. Hosoda et al. [1967] presented correlations for both frost density and frost thermal conductivity. The density of frost is correlated with the surface temperature and the air velocity. This purely empirical correlation appears limited to the conditions under which Hosoda et al. carried out their tests since it does not include the effects of air humidity and time. These factors affect the frost density according to Malhammar [1986], Sanders [1974], and Hayashi [1977]. Malhammar's correlation for frost density is semi-emperical and appears to be of most general applicability. This correlation is used in this work and is discussed in detail in Chapter 3.

From both Malhammar's and Sander's literature survey it is evident that most investigators have tried to correlate frost thermal conductivity with frost density alone, for example Hosoda [1967]. Both Sanders and Malhammar noted that a unique relationship between average thermal frost conductivity and average frost density does not exist. Malhammar noted that the discrepancies between different frost thermal conductivity correlations can be several hundred percent at low densities. Malhammar proposed a correlation for frost thermal conductivity involving a dimensionless heat resistance number. This correlation is used in this work and is discussed in section 3.4.

2.5 Heat Exchanger Models

A number of heat exchanger models have been proposed. Domanski [1982] used a tube by tube approach in modelling a wet and dry evaporator as part of a heat pump model. Domanski used the log-mean temperature difference method for a cross flow heat exchanger, and accounted for mass transfer by using the Lewis relation. Domanski also included a term for frost or condensate thickness in his overall expression for heat transfer, however he did not elaborate on the effect of the frost layer and how it affects the simulation. Tantakitti [1985] presented a heat pump model for a frosted outdoor coil. Tantakitti did not go into details on the evaporator model. The modelling of the frost layer, however, appears greatly simplified. Tantakitti used three different frost density and frost thermal conductivity values which are functions of the time only. Using this model one would expect to observe only trends, and not predictions of actual performance. Goldstein [1983] presented a complete analysis of the wet and dry plate fin tube heat exchanger. Goldstein based the heat transfer calculation on a log-mean temperature difference in the case of a dry coil and log-mean enthalpy difference in the case of a wet coil surface. Goldstein's model does not work if coil temperature drops below 32°F, and frost starts to form. Suitable required correlations, according to Goldstein, obtained in the litterature are presented as part of the model. Elmahdy and Mitalas [1977 "A Simple.."] presented a model for dry and wet heat exchangers. Elmahdy et al. approximated the cross counter flow arrangement of a multirow heat exchanger by a counter flow passage between moist air and chilled water separated by a mettalic surface. The log mean temperature method was used, by Elmahdy and Mitalas, for the dry surface, and the log mean enthalpy driving force was used for the dehumidifying coil. The air side heat transfer correlations used are correlations obtained by Elmahdy and presented in Elmahdy and Briggs [1979]. Ellison and Creswick [1978] presented an evaporator model as part of a heat pump model. Ellison et al's. model is based on the ε-Ntu method, and is therefore restricted to the dry case.

CHAPTER 3

THEORY

3.1 Heat Transfer Modes

The three basic modes of heat transfer normally considered are conductive, convective, and radiative heat transfer. At any time all these modes of heat transfer will take place in the operation of an evaporator. The radiative component is small, and can therefore be assumed negligible. The process of heat transfer through change of phase is normally grouped under the convective mode. In the operation of the evaporator, heat transfer by convection takes place on the air and refrigerant sides. Conduction is here overwhelmed by convective effects and is therefore neglected. Conductive heat transfer is of interest in the fins and in any frost layer deposited on the coil. The thermal resistance of the fins are normally accounted for by a fin efficiency. The thermal resistance of the tube wall is small enough, typically a fraction of one percent of total resistance, that it can be neglected. For mechanically expanded tube fin bonds, which is the case for the coils modelled, a contact resistance between the fin collar and the tube surface exists. This quantity is difficult to determine, but has been shown to account for as much as 5 ÷ 15%, Scheffield, Sauer, and Wood [1987], of the overall heat exchanger impedance. Figure 3.1 illustrates some of the nomenclature used in the modelling of the evaporator.

For the case of frost deposited on the coil the problem of coil performance becomes transient. The changes in coil temperatures and frost layer properties are, however, sufficiently slow so that processes can be considered quasi-static. Conditions are assumed steady at a given time step, and properties and flows calculated at the previous time step are used to determine the coil performance at the present time step.

3.2 Heat Transfer on the Air Side

The convective heat transfer is described by Newton's law of cooling. For the air side:

$$q = h_a \cdot A_o \cdot (T_a - T_{c,m}) \qquad (3.1)$$

In the case of frost or condensate on the coil the mean coil cooling temperature ($T_{c,m}$) is replaced by the mean surface temperature on the coil ($T_{s,m}$). If the coil is dry the mean surface temperature will be equal to the mean coil cooling temperature. Equations are therefore discussed with respect to the mean surface temperature. To avoid having to determine a mean surface temperature on the coil, the fin thermal efficiency is introduced. The fin efficiency is defined as the ratio of actual heat transfer to fin, (defined as heat transfer to fin at fin mean temperature) to ideal heat transfer to fin, (defined as heat transfer to fin at fin base temperature). The fin efficiency is therefore:

$$\phi = \frac{T_a - T_{f,m}}{T_a - T_{fb}} \qquad (3.2)$$

Using the fin efficiency the heat transfer equation becomes:

$$q = h_a \cdot A_t \cdot (T_a - T_{fb}) + h_a \cdot A_f \cdot \phi \cdot (T_a - T_{fb}) \qquad (3.3)$$

The fin base temperature is the tube temperature, and since wall thermal resistance is neglected it is also the tube inside wall temperature which is required for the refrigerant side calculation. In the case of condensate or frost on the coil the fin base temperature is replaced by the water film or frost surface temperature at the fin base ($T_{s,fb}$). The temperature drop across the frost layer is determined from calculated frost layer thickness and frost physical properties. The condensate layer can be either in the form of a film or drops. In either case the condensate layer is assumed thin enough so that its thermal resistance can be neglected, and therefore no temperature drop due to the wet surface will result. The fin effectiveness, ϕ_o is introduced for convenience. Equation (3.3) is then:

$$q = h_a \cdot A_o \cdot \phi_o \cdot (T_a - T_{s,fb}) \qquad (3.4)$$

$$\phi_o = (1 - \frac{A_f}{A_o} \cdot (1-\phi)) \qquad (3.5)$$

In order to determine the fin effectiveness the heat transfer coefficient h_a must be found. The results from McQuiston [1981] were found most suitable for the coils investigated. In McQuiston's paper, experimental results from his own and other investigator's work on four row coil heat transfer coefficients with varying geometry was presented graphically in a nondimensional form. McQuiston correlated the results to within ±10%. He illustrated the correlation graphically with a straight line. This straight line can be represented by equation (3.6) below. The coils used in the verification of the models proposed in this work have geometries similar to those tested, and on which the correlation is based. For the coils modelled in this work the heat transfer coefficient for a one row coil is required. Rich [1975] made tests on several coils varying only the number of tube rows. The work of Rich makes it possible to determine the heat transfer coefficient for coils with different number of rows from those on which tests are made. An equation for this is given by McQuiston [1981] and is shown below in equation (3.7). The correlation of McQuiston [1981] is:

$$j_4 = 0.2675 \cdot JP + 1.325 \times 10^{-3} \qquad (3.6)$$

where ,

$$j = St \cdot Pr^{2/3} \qquad (3.6.1)$$

$$JP = Re_D^{-0.4} \cdot (A_o / A_t)^{-0.15} \qquad (3.6.2)$$

$$\frac{A_o}{A_t} = \frac{4}{\pi}\frac{X_L}{D_h}\frac{X_T}{D_o}\,\sigma \qquad (3.6.3)$$

$$\sigma = A_{min} / A_{fc} \qquad (3.6.4)$$

$$Re_D = G_{max} \cdot D_o / \mu \qquad (3.6.5)$$

$$D_h = 4\, A_{min} \cdot d / A_o \qquad (3.6.6)$$

The tube row effect is given by:

$$\frac{j_n}{j_4} = \frac{1-1280 \cdot n \cdot Re_L^{-1.2}}{1-(4 \cdot 1280) \cdot Re_L^{-1.2}} \qquad (3.7)$$

where,

$$Re_L = G_{max} \cdot X_L / \mu \qquad (3.7.1)$$

Equations (3.6) and (3.7) give the heat transfer coefficient for the dry surface coil. Several investigators, McQuiston [1981], Myers [1967], and Eckels and Rabas [1987], noted that the heat transfer coefficient on a wetted surface is greater than on a dry surface. McQuiston's [1981] results show that this effect is more significant at higher Reynolds numbers. The source and magnitude of this enhancement is, however, not clear. The enhancement of the heat transfer coefficient on the frosted surface has also been documented. Malhammar [1986] noted that for surface temperatures below 14 °F and high relative humidity values, a considerable increase in the heat transfer coefficient for certain geometries results. He noted that insufficient results are available in order to predict the enhancement accurately. In his own tests no increase in the heat transfer coefficient was meassured on the frosted coil at temperatures above 14 °F as compared to a dry coil. He noted that under no circumstances, however, is the dry coil heat transfer coefficient greater than on the rough frosted surface. It is therefore assumed that the heat transfer coefficient on the wet and the frosted coil is the same as that on the dry coil surface. This is a conservative assumption.

$$h_{wet} = h_{dry} \tag{3.8.1}$$

$$h_{fst} = h_{dry} \tag{3.8.2}$$

The heat transfer coefficients h_{wet}, and h_{fst} are not to be confused with the added heat transfer due to latent heat transfer as a result of air dehumidification. This will be discussed in section 3.3 under mass transfer.

Having obtained the convective heat transfer coefficient the fin efficiency can be determined. Gardner solved the differential equations describing the temperature distribution in a circular fin, and presented the solution in graphical form. Gardner's results are given by Threlkeld [1970]. As is the case for most coils used in heating and air conditioning applications, the fins on the coils investigated in this work are of the flat plate type. The results for the circular fin efficiency can be used by introducing an equivalent fin diameter. This is done by calculating the diameter of a circular fin of equal area as the plate fin. Threlkeld [1970] pointed out that the validity of using an equivalent diameter was shown by Carrier and Andersson in 1944. From the graphical solution presented in Threlkeld [1970] the fin efficiency is a function of (Figure 3.2 explains nomencalture):

$$X = z_{eqv} \cdot \left[\frac{2 \cdot h_{tot}}{k_f \cdot y}\right]^{1/2} \tag{3.9}$$

$$R = \frac{D_{eqv}}{D_o} \tag{3.10}$$

$$D_{eqv} = 2 \cdot \sqrt{\frac{d_1 \cdot d_2}{\pi}} \tag{3.11}$$

It was shown by McQuiston [1975] that the fin efficiency for a wetted coil can be obtained by accounting for the added heat transfer due to mass transfer and the subsequent phase change of the water vapor. The heat transfer coefficient in equation (3.9) for the dehumidifying coil is therefore:

$$h_{tot} = h_a + h_{lat}. \qquad (3.12)$$

The derivation of h_{lat} is explained in section 3.3 below. When the air is dehumidified the effect of the increased total heat transfer coefficient is a decrease in the fin efficiency. This can be seen from the fin efficiency curves of Threlkeld [1970] considering an increase in the heat transfer coefficient, other parameters being equal. If the coil temperature is below the freezing point of water the accumulation of frost on the coil will have an insulating effect on the fins. The result is an increase in the fin efficiency. There are, however few studies available on this topic, and those that are availble are analytical in nature. Kondepudi [1987] reviewed the work in this area. Since no experimentation has been done to verify any of the analytical work done in this area, and the work lacks in rigour, the insulating effect and possible increase in fin efficiency is assumed negligible. To use the results of Gardner in a numerical model the fin effectiveness curves need to be written in the form of equations. The equations of Tantakitti [1985] are:

$R = 1.0$ (3.13.1)

$$\phi = 1.03552448 - 0.31837607 \cdot X + 0.02589744 \cdot X^2 - 0.00101 \cdot X^3$$

$R = 1.5$ (3.13.2)

$$\phi = 1.04188811 - 0.3713209 \cdot X + 0.04282051 \cdot X^2 - 0.000637 \cdot X^3$$

$R = 2.0$ (3.13.3)

$$\phi = 1.03552448 - 0.45592852 \cdot X + 0.08051282 \cdot X^2 - 0.005004 \cdot X^3$$

$R = 3.0$ (3.13.4)

$$\phi = 1.03881119 - 0.5195338 \cdot X + 0.10275058 \cdot X^2 - 0.0070863 \cdot X^3$$

$R = 4.0$ (3.13.5)

$$\phi = 1.03230769 - 0.5540948 \cdot X + 0.117016 \cdot X^2 - 0.0087024 \cdot X^3$$

Linear interpolation is used for values of R, that fall in between those represented by the cubics above.

3.3 Mass Transfer

When the coil surface temperature falls below the saturation temperature corresponding to the air partial vapor pressure the air will be dehumidified. Water vapor will be transferred towards the coil surface, in direction of the negative partial vapor pressure gradient, and deposit as either water or frost on the coil surface. The temperature corresponding to saturation at the air partial vapor pressure is called the dew point. If the coil surface temperature is below the dew point and above 32 °F water vapor will condense on the coil. If the coil temperature is below the dew point and 32 °F frost will deposit on the coil surface. The process of air dehumidification and the associated heat flows on a section of a frosted coil surface is illustrated in Figure 3.3.

The effect of air dehumidification is an increase in heat transfer. The source of the increased heat transfer is the energy released in the change of phase of the water vapor. The rate of water vapor flow towards the coil surface is proportional to the vapor concentration gradient and is given by:

$$
\begin{aligned}
m_v &= \beta \cdot A_o \cdot (\rho_{va} - \rho_{vs,m}) \\
&= (\rho_{da} \cdot \beta) \cdot A_o \cdot \frac{(\rho_{va} - \rho_{vs,m})}{\rho_{da}} \\
&= h_m \cdot A_o \cdot (w_a - w_{s,m}) \qquad (3.14)
\end{aligned}
$$

The convective mass transfer is related to the convective heat transfer coefficient by the Lewis number as defined in Threlkeld [1970]. The Lewis number is then given by:

$$Le = \frac{h_a}{h_m \cdot C_{p,da}} \qquad (3.15)$$

In more recent literature, for example ASHRAE [1981], the Lewis number is defined as the ratio of thermal diffusivity to mass diffusivity. The end result is independent of the definition selected. With Lewis number defined by equation (3.15) the latent heat transfer due to air dehumidification is expressed as:

$$q_{lat} = \frac{h_a}{Le \cdot C_{p,da}} \cdot A_o \cdot i_w \cdot (w_a - w_{s,m}) \qquad (3.16)$$

As was the case for the sensible heat transfer to the finned coil it is desirable to express the latent heat transfer in terms of the humidity at saturation corresponding to the surface temperature at the fin base. The following steps will allow this:

$$q_{lat} = h_m \cdot A_f \cdot i_w \cdot (w_a - w_{s,f,m}) + h_m \cdot A_i \cdot i_w \cdot (w_a - w_{s,fb})$$

$$= h_m \cdot i_w \cdot (w_a - w_{s,fb}) \cdot (\hat{\phi}\, A_f + A_i) \qquad (3.17)$$

where,

$$\hat{\phi} = \frac{w_a - w_{s,f,m}}{w_a - w_{s,fb}} \qquad (3.18)$$

It is assumed that,

$$\hat{\phi} = \phi \qquad (3.19)$$

Equation (3.17) is therefore:

$$q_{lat} = h_m \cdot i_w \cdot A_o \cdot \phi_o \cdot (w_a - w_{s,fb}) \qquad (3.20)$$

The total heat transfer, latent and sensible, can be expressed as:

$$q_{tot} = q_{sen} + q_{lat} \qquad (3.21)$$

$$q_{tot} = h_a \cdot A_o \cdot \phi_o \cdot (T_a - T_{s,fb}) + h_a \cdot \frac{i_w}{Le \cdot C_{p,da}} \cdot \phi_o \cdot A_o \cdot (w_a - w_{s,fb})$$

$$= h_a \cdot A_o \cdot \phi_o \cdot \left[1 + \frac{i_w}{Le \cdot C_{p,da}} \cdot \left(\frac{w_a - w_{s,fb}}{T_a - T_{s,fb}}\right)\right] \cdot (T_a - T_{s,fb}) \qquad (3.22)$$

let,

$$C = \frac{W_a - W_{s,fb}}{T_a - T_{s,fb}} \tag{3.23}$$

then,

$$q_{tot} = h_a \cdot \left[1 + \frac{i_w \cdot C}{Le \cdot C_{p,da}}\right] \cdot A_o \cdot \phi_o \cdot (T_a - T_{s,fb}) \tag{3.24}$$

which can be written as,

$$q_{tot} = (h_{sen} + h_{lat}) \cdot A_o \cdot \phi_o \cdot (T_a - T_{s,fb}) \tag{3.25}$$

where,

$$h_{sen} = h_a \tag{3.26}$$

$$h_{lat} = \frac{h_a \cdot i_w \cdot C}{Le \cdot C_{p,da}} \tag{3.27}$$

These equations have the important feature that in the case of no air dehumidification, the parameter C is zero and equation (3.25) reduces to the dry air side equation. The value of C is obtained by iterations and the method is explained in Chapter 4. Using the information in Chapter 10 of Trelkeld [1970] a value of Lewis number in the region 0.90 to 0.92 is obtained. Most authors that have made use of the Lewis number in modelling the mass transfer on cooling coils, Domanski [1982], McQuiston [1975] and Malhammar [1986], used a Lewis number of 1, without justifying this selection. In the numerical modelling of the evaporators used to verify the models proposed in this work different Lewis numbers between 0.91 and 1.0 were used. A value of 0.95 was finally choosen, since this appeared to predict air dehumidification and heat transfer rate most accurately.

3.4 The Growth of the Frost Layer

As mentioned when the surface temperature of the coil falls below the dew point and the freezing point simultaneously, desublimation of vapor in air will take place and frost will deposit on the coil surface. Malhammar [1986] identifies three forms of frost formation.

1. Water vapor diffuses towards the surface and is deposited as frost.
2. Water vapor diffuses towards the cooling frost surface which is at 32 °F. The vapor will therefore fall out as water. This water is absorbed in the frost layer and freezes to ice deeper in the layer. As a result the thermal resistance of the frost layer is decreased, leaving the surface temperature below 32 °F . After some additional frost is built up the whole process is repeated.
3. In the case of frost particles in the air some will collide and stick to the surface.

Cases 2 and 3 are extremely difficult to model. Malhammar's frost property correlations, used in this work, are restricted to case 1.

The accumulation of frost on the coil introduces considerable complications in the coil modelling. The coil performance is now a transient phenomena. The growth of the frost layer increases thermal resistance, and reduces free flow through area. As a result heat transfer, and refrigerant and air mass flow rates are affected. These transients, however, are small enough that a quasi static state can be assumed for the coil at all times. To determine the effect of the frost layer on coil performance it is crucial to have good estimates of the frost layer density and thermal conductivity.

Malhammar [1986] noted that prediction accuracy of the frost growth depends mainly on the frost density. The most influential parameters on frost density, according to Malhammar's work appear to be: time, coil cooling temperature, air velocity, and air humidity. Malhammar noted that during the first stage of the frost growth a high density layer is formed. The layer then grows towards a decreasing density. This period is short. In the second growth period the frost density increases with time. He also noted that a decrease in the coil cooling temperature has the effect of decreasing frost density, and a high air velocity gives a denser frost. Malhammar pointed out that an increased air velocity also has the effect of increasing the heat transfer coefficient which results in an increased water vapor flow towards the surface. These two effects more or less cancel each other, and as a result the effect of the air velocity on the frost layer thickness is small. Malhammar claimed that when the air relative humidity is low the outside diffusion of water vapor will decrease in relation to the diffusion in the frost layer, which takes place at saturation. As a result the density of the frost will increase. Malhammar took account of all these factors in his semi-empiical correlation for frost density, in the form of dimensionless numbers. Of all the frost property correlations available in the literature the semi-empirical correlations of Malhammar appear to be of most general applicability. Malhammar's equations are in SI units. The equations are:

$$Ft = Vm\cdot(1.05 + \sqrt{0.693 + K_1\cdot Ti}\ ) \tag{3.28}$$

where Ti, the time number is given by:

$$Ti = \frac{t\cdot i_{sg}\cdot q_{lat}}{q_{tot}\left[\frac{D\cdot i_{sg}}{R_v\cdot(T_{c,m}+273)}\right]} \tag{3.29}$$

The frost number, Ft is:

$$Ft = \frac{\rho_{fst}\cdot R_v}{\left[\frac{q_{tot}}{q_{lat}}\cdot\left(\frac{dP_v}{dT_{c,m}}\right)^{*}\right]} \tag{3.30}$$

$$\frac{D\cdot i_{sg}}{R_v\cdot(T_{c,m}+273)}^{*} = 4.98\times10^{-4}\cdot(1+T_{c,m}/396) \tag{3.31}$$

where $T_{c,m}$ is the mean coil cooling temperature in units of °C.

The rate of change of saturated water vapor pressure with temperature is given by:

$$\left[\frac{dP_v}{dT_{c,m}}\right]^n = 4.325\times10^{10}\cdot\exp\left(\frac{-5619}{273+T_{c,m}}\right) \qquad (3.32)$$

The heat resistance number V_m, and the empirical quantity K_1 are obtained from the following schedule:

$0\leq Re_o\leq 2600$	$Vm=204$	$K_1=2.58\times10^{-14}+1.91\times10^{-16}\cdot Re_o$
$2600\leq Re_o\leq 5000$	$Vm=113+3.50\times10^{-2}\cdot Re_o$	$K_1=5.23\times10^{-10}$
$5200\leq Re_o\leq 22000$	$Vm=295$	$K_1=3.06\times10^{-2}\cdot Ft_{ice}^{1.90}$

(3.33)

Reynolds number is calculated from:

$$Re_o = \frac{V_{loc}\cdot2\cdot s_o\cdot\rho_a}{\mu_a} \qquad (3.34)$$

and the local velocity is defined as (see Figure 3.4):

$$V_{loc} = V_{fc}\cdot\frac{s_o+y}{s} \qquad (3.35)$$

$$s = s_o - 2\cdot\delta_{fst} \qquad (3.36)$$

and the ice frost number:

$$Ft_{ice} = Ft\cdot\frac{\rho_{ice}}{\rho_{fst}} \qquad (3.37)$$

From these calculations the frost number can be determined using equation (3.28). From the definition of frost number the frost layer density is calculated :

$$\rho_{fst} = \frac{Ft\cdot\left[\frac{q_{tot}}{q_{lat}}\cdot\left(\frac{dP_v}{dT_{c,m}}\right)^n\right]}{R_v} \qquad (3.38)$$

Except for a few points all Malhammar's test points were within ±20% of those predicted from equations above.

Equations correlating frost thermal conductivity with frost density have been proposed by several investigators. Malhammar illustrated the large discrepancies between the different correlations. At low densities he noted that the predicted conductivity values may differ by several hundred percent, and noted that the heat conductivity can not be a function of frost density alone. Malhammar proposed that the frost layer heat conductivity be calculated from:

$$k_{fst} = \frac{0.202 \cdot \rho_{fst} \cdot (1 - \rho_{fst}/1860)}{Vm - 0.189 \cdot \rho_{fst}} \tag{3.39}$$

Where k_{fst} is in units of (W/m·C), and the frost density has units of (Kg/m^3). The heat resistance number Vm, is given from equations (3.33).

The frost density and conductivity calculated above are average values for the frost layer. Since surface temperatures, ratio of latent to sensible heat transfer rates, and air velocities are all varying with time, time averages must be calculated for all relevant parameters in order to obtain the average properties of the frost layer at a given time. The following averaging formulas have, according to Malhammar, been shown to give good results. The time average Reynolds number is calculated from:

$$Re_{avg} = \frac{1}{2} (Re_m + Re_o) \tag{3.40}$$

where

$$Re_m = \frac{2 \cdot s_o}{t} \int_0^t (V_{tot} \cdot \rho_a / \mu_a) \, d\tau \tag{3.41}$$

$$\cong \frac{\Delta t}{t} \sum_i (Re_o)_i \tag{3.42}$$

In a similar manner:

$$\left[\frac{q_{tot}}{q_{lat}} \cdot \left(\frac{dP_v}{dT_{c,m}}\right)^m \right]_{avg} \cong \frac{\Delta t}{t} \sum_i \left[\frac{q_{tot}}{q_{lat}} \cdot \left(\frac{dP_v}{dT_{c,m}}\right)^m \right]_i \tag{3.43}$$

$$\left[\frac{q_{lat}}{q_{tot}}\right]_{avg} \cong \frac{\Delta t}{t}\sum_i \left[\frac{q_{lat}}{q_{tot}}\right]_i \qquad (3.44)$$

The time average of the mean coil cooling temperature is:

$$(T_{c,m})_{avg} \cong \frac{\Delta t}{t}\sum_i (T_{c,m})_i \qquad (3.45)$$

These time averages are then used to calculate the required quantities to find Ft, the frost number, from equation (3.28) and subsequently the average frost density and conductivity from equations (3.38) and (3.39) respectively.

Equation (3.28) is applicable for air and atmospheric pressure with ±25% accuracy in the following ranges:

Ft	400 - 3000	
$\frac{q_{tot}}{q_{lat}}\cdot(\frac{dP_v}{dT_{c,m}})^{"}$	40 - 300	(Pa/°C)
Ti	0.75×10^{10} - 5.5×10^{10}	
Re_o	0 - 5200	(Re_0 > 5200 unpredictable)
ρ_{fst}	90-450	(Kg/m^3)
$T_{c,m}$	-3 - -25	(°C)
$T_{ai}-T_{c,m}$	2.5 - 14	(°C)
φ	60 - 90	(%)
s_o	3.2 - 15.6	(mm)

From the calculated frost density and thermal conductivity the frost layer thermal resistance can be determined. The frost layer thickness is determined from:

$$\delta_{fst} = \frac{m_{fst}}{A_o\cdot\rho_{fst}} \qquad (3.46)$$

and the heat transfer through the frost layer is therefore:

$$q_{tot} \cong \frac{k_{fst}\cdot A_o}{\delta_{fst}}(T_{s,fb}-T_{fb}) \qquad (3.47)$$

3.5 Heat Transfer on the Refrigerant Side

The refrigerant absorbs energy in the two phase state or the superheated state. The refrigerant side two phase heat transfer coefficient is an order of magnitude larger than the heat transfer coefficient for the superheated refrigerant. The evaporator thus performs best when the refrigerant is wet at all times. Superheat is sometimes required to prevent liquid entering the compressor in the heat pump system. The numerous correlations proposed for the heat transfer coefficients of boiling liquids suggests the complicated nature of the forced convection evaporation heat transfer process.

The heat transfer process on the refrigerant side can be considered in three different regions. The first is the two-phase region, the second is the transition region, and the third is the superheated region. In the two phase region Pierre's correlation [1955] was found to best suit the conditions under which the evaporator coils modelled in this work were operated. The superheated refrigerant heat transfer coefficient is calculated from a correlation for turbulent single phase flow suggested by Hiller [1976]. The heat transfer coefficient in the transition region is interpolated between the two phase heat transfer coefficient corresponding to the inlet quality of the transition region, and the superheated heat transfer coefficient value coresponding to the saturated vapor refrigerant, quality equal to 1, leaving the transition region.

Pierre's correlation applies for incomplete evaporation from an inlet quality of about 0.2, and an exit quality of 0.9. Several investigators have noted that as quality approaches unity, dry patches appear on the heated surface thus reducing the heat transfer, Anderson, Rich, and Geary [1966]. The transition region according to Pierre's correlation should then be between an inlet quality of 0.9 and leaving at unity. It is noted by Nussbaum in

Polke [1963] that findings by early investigators indicate that a laminar flow pattern with little wetting occurs at low rates of heat flux density. Definite separation of liquid and gas at low heat flux densities, often encountered in air cooling units, had been noted. The non wetted surfaces participate only to a small extent reducing the heat transfer coefficient in such areas considerably. The heat flux densities for the coils investigated in this work were very low. Use of a transition region between qualities of 0.9 and 1 in the numerical model gave consistent overestimates of the heat transfer rate to the refrigerant as compared to the experimental results. It was therefore logical to consider an extended transition region. There were no means of establishing a region of refrigerant dry-out, but a transition region with an inlet quality of 0.75 and saturated vapor leaving, improved model predictions considerably. The transition region for the coils investigated was therefore selected in the refrigerant quality range 0.75 to 1.0. This might not be applied in general but is one variable that needs attention in evaporator modelling.

Pierre's correlation for the evaporation of Refrigerants 12, 22, and 11 takes the form:

$$\frac{h_{tp} \cdot D_i}{k_{r,l}} = 0.0009 \cdot \left[Re_r^2 \cdot Kf \right]^{0.5} \tag{3.48}$$

$$Re_r = \frac{G_r \cdot D_i}{\mu_{r,l}} \tag{3.48.1}$$

$$Kf = \frac{\Delta x \cdot i_{r,fg}}{\Delta L \cdot g} \tag{3.48.2}$$

$$10^9 < Re_r^2 \cdot Kf < 0.7 \times 10^{12}$$

where Kf is the boiling number. Pierre did not elaborate on the physical significance of the boiling number. The boiling number appears, however, to be a nondimensionalized meassure of the heat addition per unit length on the refrigerant side. It is therefore clear that the refrigerant heat transfer coefficient will depend on the air side conductance. A lower air side heat transfer coefficient will reduce the refrigerant side heat transfer coefficient.

The heat transfer coefficient of turbulent superheated refrigerant flow is given by the following equation suggested by Hiller [1976]:

$$St \cdot Pr^{2/3} = 0.0108 \cdot Re^{-0.1375}$$

$$\Rightarrow \quad h_{sup} = 0.0108\, G_r \cdot C_{p,r}^{1/3} \cdot k_r^{2/3} \cdot \mu_r^{-2/3} \cdot \left(\frac{G \cdot D_i}{\mu_r}\right)^{-0.1375} \qquad (3.49)$$

$$Re > 6000$$

The refrigerant heat transfer coefficient in the transition region is obtained by interpolating the two phase and the superheated values. The following smooth interpolation function is used.

$$h_{trn} = h_{2p} \cdot \sin^2\left(\frac{\pi}{2}(1-\omega)\right) + h_{sup} \cdot \cos^2\left(\frac{\pi}{2}(1-\omega)\right) \qquad (3.50)$$

where,

$$\omega = \frac{x - x_o}{1 - x_o} \qquad (3.50.1)$$

and x_o is the inlet quality to the transition region. Figure 3.5 illustrates the heat transfer coefficient as a function of refrigerant quality in the transition region.

The local heat transfer on the refrigerant side can therefore be written as:

$$q = h_r \cdot A_i \cdot (T_{fb} - T_r) \qquad (3.51)$$

where, $h_r = h_{2p}$, in the two phase region

$h_r = h_{trn}$, in the transition region

$h_r = h_{sup}$, in the superheated region

3.6 Contact Resistance and Fouling of Surfaces

Contact resistance arises when heat is transfered across an interface were two surfaces make imperfect contact. The most common means of bonding the fins to the tubes on heat pump coils is by mechanical expansion. This bond is imperfect and a meassurable temperature drop across the bonding surface results. The problem of estimating the thermal contact conductance is a difficult one, and has attracted the attention of many researchers. One of the more recent investigations is by Sheffield, Abu-Ebid, and Sauer [1985] who presented a correlation of the dry contact conductance for mechanically expanded tube-fin bonds, in terms of fin collar length in contact with the tube, fin thickness, net interference, and unexpanded tube diameter. The correlation is:

$$h_c = 4.957\times10^{22}\cdot\left[\left(\frac{y}{D_o}\right)^{10.005}\cdot\left(\frac{I+0.0205}{L_c}\right)^{2.967}\right] \quad (3.52)$$

where, I the net interference is given by:

$$I = (d_e + 2\cdot\delta_w - d_{fc}) \quad (3.53)$$

This contact conductance is expressed in terms of the contact area between the fin collar and the tube. Based on scanning electron microscope photographs, for a coil with fin density 14 fpi, Sheffield et al. calculated the actual collar length in contact with the tubes using, (1/fpi - 2y). Sheffield et al. [1985] observed that increasing the fin density from 14 to 20 fins per inch, increased the average contact conductance value from 1868 to 4087 Btu/(hr ft^2).

For the coils used in the verification of the models proposed in this thesis it was not possible to determine the fin interference, I, due to lack of information. Also the contact area was unknown. From the results of Sheffield et al. [1985], the contact area is approximately 90% for a 14 fpi coil. To investigate the effect of the contact conductance on coil performance a value of 2000 Btu/(hr °F ft^2) for the contact conductance and 100% contact area for the 8 fpi coils used in the verification of the models in this thesis, were assumed.

$$q = h_c \cdot A_t \cdot \Delta T_c \tag{3.54}$$

$$h_c = 2000 \text{ Btu/(hr}\cdot\text{F ft}^2) \tag{3.54.1}$$

Wood, Sheffield, and Sauer ["Effect of" 1987] concluded in a more recent paper that contact conductance does not significantly affect the heat transfer. This was confirmed by numerical simulations using the assumed 2000 (Btu/hr•F ft^2) and 100% contact area.

When the evaporator has been in use for an extended period of time the heat transfer surfaces on both the air, and refrigerant sides become fouled up. Two coils were used in the verification of the numerical models. One coil had been in operation for five years, and the second coil was purchased in February 1988, when tests were run on the new coil. The compressor oil is soluble in the refrigerant, and is therefore carried with the flow throughout the heat pump system. Some oil will circulate in the system even if an oil separator is used. As the refrigerant is vaporized a small film of oil is left behind. This film was observed in the piping of an evaporator that had been in use for an extended period of time. In the modelling of the evaporator that had been in use for five years, better agreement with experimental results was observed if a fouling resistance factor was included. For the new, and clean coil better agreement was observed when no fouling on neither air nor refrigerant side was assumed. Rosenhow [1985] gave a fouling factor for oil bearing refrigerant vapors of 500 Btu/(hr•F ft^2), and for operation in industrial air 500 Btu/(hr•F ft^2) as an air side fouling factor. These fouling factor values were used in the numerical models for the five year old evaporator coil.

$$h_{f,a} = 500 \text{ Btu/(hr}\cdot\text{F ft}^2) \tag{3.55.1}$$

$$h_{f,r} = 500 \text{ Btu/(hr}\cdot\text{F ft}^2) \tag{3.55.2}$$

The industrial air assumption can be questioned. The predictions of the heat transfer rate of the numerical model were on the high side even after introduction of the refrigerant fouling factor. The air side fouling factor improved the agreement, but only a slight change was evident since the air side contact resistance is applied on the outside finned surface, which is an order of magnitude greater than the inside surface area. The coil has operated in a city environment with no filters for five years so the presence of fouling on the air side to some degree is obvious.

3.7 The Complete Heat Transfer Equation

The local heat transfer to the evaporator can now be conveniently expressed as a function of the temperature difference between the air and the refrigerant sides. Combining equations (3.24), (3.47), (3.51), (3.54), and including the fouling factors (3.55.1) and (3.55.2) the heat transfer equation becomes:

$$q = \frac{(T_a - T_r)}{\dfrac{1}{h_a\left[1+\dfrac{i_w \cdot C}{Le \cdot C_{p,da}}\right]A_o \cdot \phi_o} + \dfrac{\delta_{fst}}{k_{fst} \cdot A_o} + \dfrac{1}{h_{f,a} \cdot A_o} + \dfrac{1}{h_{f,r} \cdot A_i} + \dfrac{1}{h_c \cdot A_i} + \dfrac{1}{h_r \cdot A_i}} \qquad (3.56)$$

3.8 Pressure Drop on the Air side

The air flow across the coil is determined from the intersection of the fan characteristic (pressure head a monotonically decreasing function of air flow rate) and the coil pressure drop characteristic (pressure drop a monotonically increasing function of air flow rate). The two major components of air side pressure drop are form drag pressure loss and skin friction pressure loss. The acceleration term is negligible for heat pump evaporator coil applications. The pressure loss is normally expressed in terms of a friction factor. Since form drag accounts for a significant portion of the pressure loss the term friction factor can be misleading. Rich [1973] noted that the form drag is produced mostly by the tubes, and the fins are the main source of skin friction. He then separates the friction factor into a tube, and a fin component. Most investigators, however, group the effect of tubes and fins in one friction factor. Given the Fanning friction factor, f, the pressure drop is normally given by:

$$\Delta P_{air} = f \cdot \frac{d}{D_h} \cdot \frac{G^2_{max}}{2\rho} \tag{3.57}$$

$$D_h = 4 \cdot A_{min} \cdot d / A_o \tag{3.57.1}$$

Several researchers have presented correlations for the friction factor. The correlations of McQuiston [1981], and Turaga, Lin, and Fazio [1988] appear to be of most general applicability.

In the numerical models the air flow for the dry and wet coil is not calculated but is a required input. The input is either the air volume flow rate or the initial non frosted coil pressure drop. The model could easily be modified so that the initial air volume flow rate is calculated if the fan characteristic is available and a correlation representative of the coil pressure drop is located. One of the objectives for the use of this model is to study the influence of various input parameters on heat pump performance. The air volume flow rate is one such parameter. Rather than having to use different fan characteristics to obtain different air flow rates, it is more convenient to use the air volume flow, or initial pressure drop as an input variable.

As frost is deposited on the coil the pressure drop across the coil will increase as a result of reduced flow through area and increased surface roughness. Malhammar [1986] calculated the pressure drop for a frosted coil accounting for the reduced flow area, but using a correlation based upon a smooth surface. He then plots this pressure drop against time together with experimental results for pressure drop on the same coil. From the plot, Malhammar noted that it is evident that the pressure drop is due to the surface roughness almost exclusively. In order to model the frosting process properly the fan characteristic together with a correlation for pressure drop as a function of frost deposition is required. Malhammar noted that the meassuring accuracy in the quantities required for friction factor determination is poor. In calculating the friction factor he notes that for a 20% error in the meassured free flow area, the friction factor, compared to the correct value will vary in the range 50–170%. As a result, Malhammar noted that a friction factor correlation for the frosted case must be rather crude. From his own experimental results Malhammar suggested that the friction factor is represented rather well by the greater of:

$$f = f_{unfrosted} \tag{3.58.1}$$

$$f = k_1 \cdot \delta_{fst} \cdot \rho_{fst} \tag{3.58.2}$$

where,

$$k_1 = 2.5 \ m^2 / Kg \tag{3.58.3}$$

$$= 12.23 \ ft^2 / lb_m$$

The calculation of the unfrosted factor should, according to Malhammar, be based on the free flow area. He noted that the term $\delta_{fst} \cdot \rho_{fst}$ represents the amount of frost per unit surface area. Using the friction factor given by (3.58.2) the pressure drop, according to Malhammar, is given by

$$\Delta P_{air} = f \cdot \frac{d}{d_h} \cdot \rho_a \cdot \frac{V_{loc}^2}{2} \tag{3.59}$$

where V_{loc} is determined from equation (3.35), and

$$d_h = 2 \cdot s \tag{3.59.1}$$

where s is determined using equation (3.36).

In the models in this thesis the frosted friction factor in equation (3.58.2) is used when the pressure drop calculated using equation (3.59) is greater than the initial user supplied pressure drop.

3.9 Pressure Drop on the Refrigerant Side

The pressure drop on the refrigerant side is composed of frictional losses and losses due to flow acceleration. The superheated refrigerant pressure loss calculation presents little problem. The pressure loss in the two-phase region is, however, more complicated. Assuming homogeneous flow allows for a physically meaningful derivation of the pressure loss equations for friction and acceleration. In homogenous flow the two-phase mixture is treated as a pseudo fluid that obeys the usual equations of single component flow. Suitable average properties are then determined and used in the equations. For two-phase flow in which one phase is finely dispersed in another, homogeneous flow, Wallis [1969] suggests the following equation to determine the pressure drop:

for friction loss,

$$-\left(\frac{dP_r}{dL}\right)_{fr,tp} = \frac{2 \cdot f_r \cdot G_r^2}{D_i}\left(v_f + x \cdot v_{fg}\right) \qquad (3.60)$$

where,

$$f_r = 0.079 \left(\frac{1}{Re_r}\right)^{0.25} \qquad (3.60.1)$$

and,

$$Re_r = \frac{G_r \cdot D_i}{\mu_r} \qquad (3.60.2)$$

$$\mu_r = x \cdot \mu_g + (1 - x) \cdot \mu_f \qquad (3.60.3)$$

The acceleration pressure drop is obtained by considering the one dimensional momentum equation. The pressure drop due to flow acceleration in a constant area section is given by:

$$-\left(\frac{dP_r}{dL}\right)_{acc} = G_r^2 \cdot \frac{dv}{dL} \qquad (3.61)$$

The friction pressure drop for the superheated refrigerant can be determined using the Darcy-Weisbach relation, ASHRAE Fundamentals [1981]. The equation is:

$$-\left(\frac{dP_r}{dL}\right)_{fr,s} = f \cdot \frac{1}{D_i} \cdot \frac{G_r^2}{2\,\rho_r} \qquad (3.62)$$

where,

$$f = 0.3614 \cdot Re_r^{-0.25}\ ,\quad Re_r < 10^5 \qquad (3.62.1)$$

$$f = 0.0032 + 0.221 \cdot Re_r^{-0.237}\ ,\quad 10^5 < Re_r < 3\times10^6 \qquad (3.62.2)$$

The pressure drop due to flow acceleration is negligible in the superheated region.

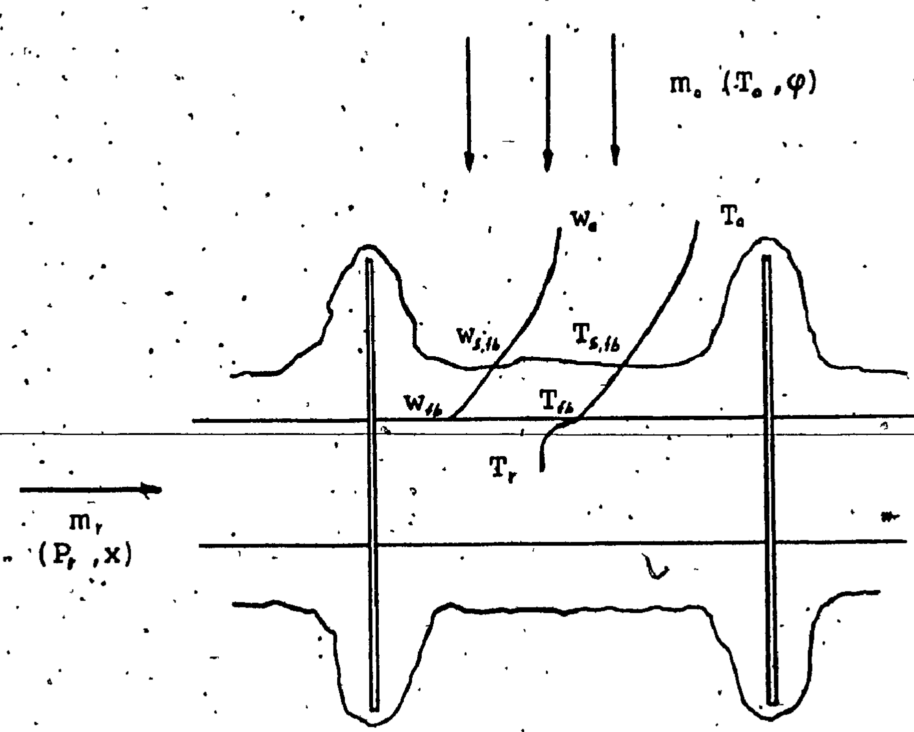


FIGURE 3.1

Section of Frosted Finned Tube Evaporator

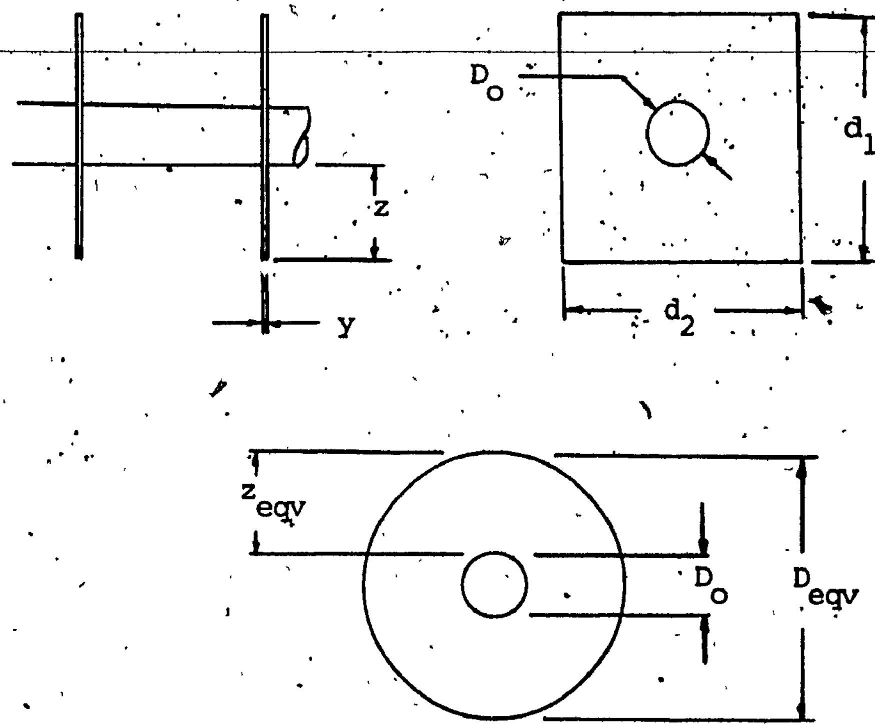


FIGURE 3.2

Fin Efficiency Nomenclature

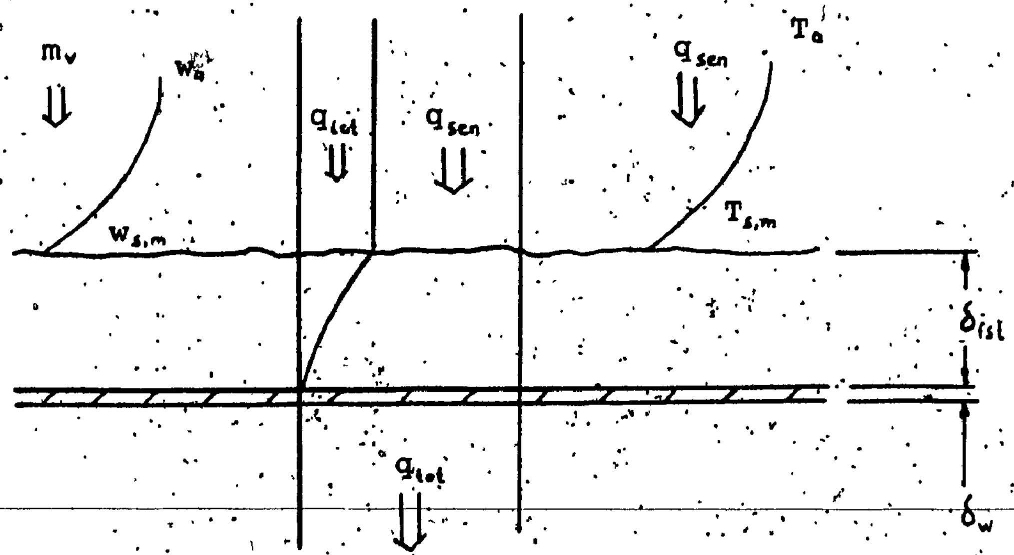


FIGURE 3.3

Heat Flows on a Frosted Coil Section

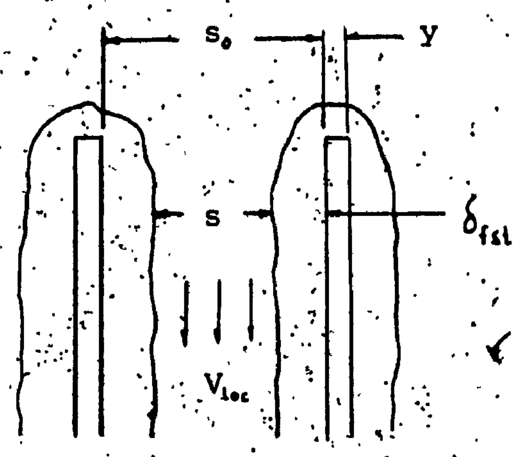


FIGURE 3.4

Local Air Velocity

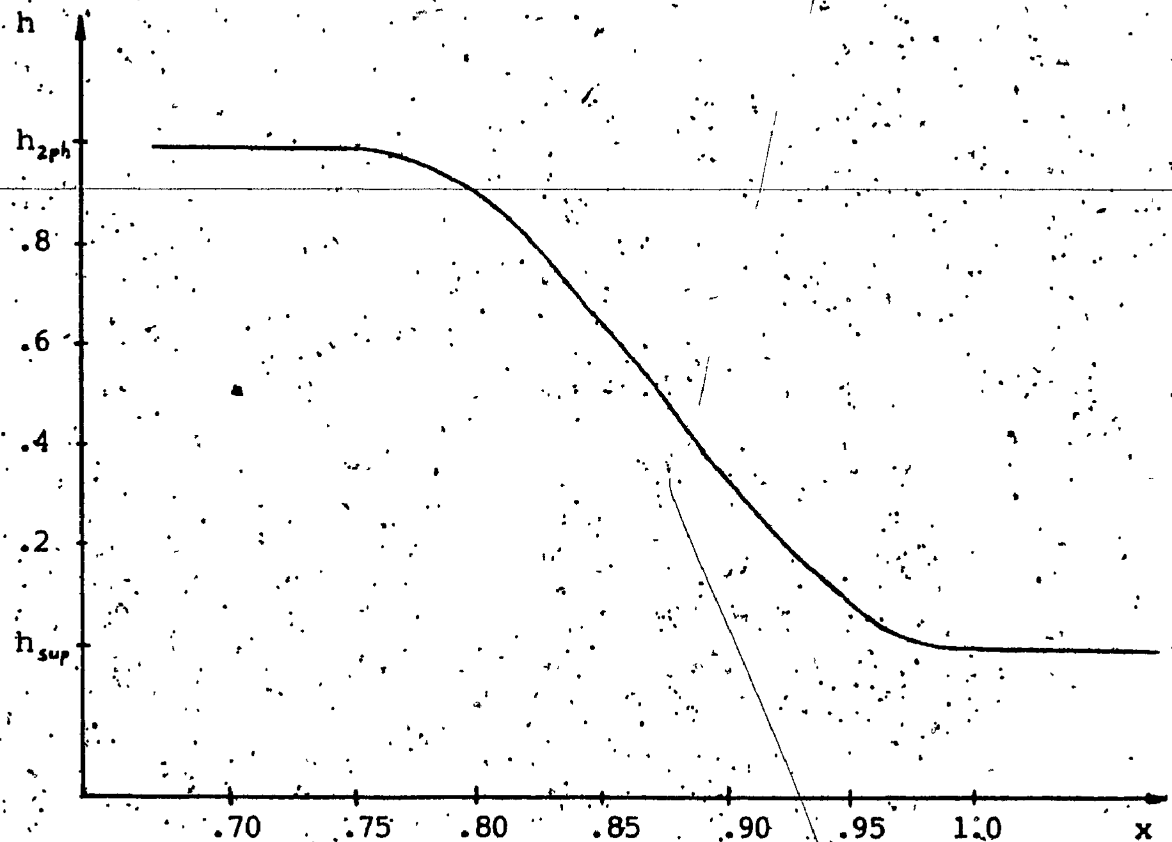


FIGURE 3.5

Transition Region Heat Transfer Coefficient

CHAPTER 4
THE EVAPORATOR MODELS

4.1 The Simulation Method

The evaporator models are designed to simulate coil performance sufficiently long after the start up period so that related transients are negligible. In the case of a dry and wet coil surface the operation will be steady state; however, for frost forming on the finned plate heat exchanger, the simulation will be of a transient nature. The time variation, however, is sufficiently small that a quasi-steady state can be assumed for the coil frosting process with good accuracy. The time step is assumed short enough so that parameters calculated at time t can be stored and used in the calculations at time t+Δt.

Although the finite element and the three region models are different, the approach to modelling of the multi-row evaporator is similar. A general multi-row coil with a number of tube passes in each circuit is illustrated in Figure 4.1. It is assumed in the models that each circuit is equivalent to one row. Hence in this discussion rows are equivalent to circuits. The multi-row coil is modelled by considering a one row coil representative of an average row, subject to average air temperature, and average dehumidification rates. The air side heat transfer to one row, assuming a constant surface temperature, is given by equation (3.25). Assuming a constant heat transfer coefficient in the coil, the sensible heat transfer to n rows, all at a constant surface temperature, is therefore given by:

$$q_{sen} = h_a \cdot A_o \cdot n \cdot \phi_o \cdot (T_{av} - T_{s,fb}) \qquad (4.1)$$

where T_{av} is the average air temperature as the air stream is forced through the coil. The latent heat transfer for the n-row coil is described in a similar way:

$$q_{lat} = h_m \cdot i \cdot A_o \cdot n \cdot \phi_o \cdot (w_{av} - w_{s,fb}) \qquad (4.2)$$

where w_{av} is the specific humidity representetive of the average condition at the average temperature.

The total heat transfer (latent plus sensible) is then:

$$q_{tot} = h_a \cdot \left[1 + \frac{i_w \cdot C}{Le \cdot C_{p,da}}\right] \cdot A_o \cdot n \cdot \phi_o \cdot (T_{av} - T_{s,fb}) \qquad (4.3)$$

where the parameter C , for the average row, is given by:

$$C = \frac{w_{av} - w_{s,fb}}{T_{av} - T_{s,fb}} \qquad (4.4)$$

The air side thermal resistance for the n-row cuil as represented by the average row is therefore:

$$Res_a = \frac{1}{(h_a + h_{lat}) \cdot A_o \cdot n \cdot \phi_o} \qquad (4.5)$$

where h_{lat} is given by equation (3.27). Similar equations are easily obtained for the frost layer, outside and inside fouling, contact, and refrigerant thermal resistances by multiplying the heat transfer areas by the number of tube rows (all area calculations are based on one row):

$$Res_{fst} = \frac{\delta_{fst}}{k_{fst} \cdot A_o \cdot n} \qquad (4.6)$$

$$Res_{f,a} = \frac{1}{h_{f,a} \cdot A_o \cdot n} \qquad (4.7)$$

$$Res_c = \frac{1}{h_c \cdot A_i \cdot n} \qquad (4.8)$$

$$Res_{f,r} = \frac{1}{h_{f,r} \cdot A_i \cdot n} \qquad (4.9)$$

$$Res_r = \frac{1}{h_r \cdot A_i \cdot n} \qquad (4.10)$$

The heat transfered to the n-row coil as represented by the average row is therefore given by:

$$q = \frac{(T_{av} - T_r)}{\Sigma \, Res_i} \qquad (4.11)$$

The temperature difference, $T_{av} - T_r$, is the log-mean temperature difference between the air stream and the refrigerant for a cross flow heat exchanger. The log-mean temperature difference for a cross flow heat exchanger with constant refrigerant temperature is, according to Kays and London [1984], given by:

$$\Delta T_{lm} = \frac{T_{a1} - T_{a2}}{\ln\left[\frac{T_{a1} - T_r}{T_{a2} - T_r}\right]} \qquad (4.12)$$

This log-mean equation accounts for the air temperature variation across the coil, and applies only when the refrigerant temperature is constant. The refrigerant temperature will change if heat is transferred to it in the superheated state, and if there is significant pressure drop in the saturated state. As will be shown below, for the models developed, equation (4.12) can be used almost exclusively.

The surface temperature calculated at the average row will be slightly higher than the lowest surface temperature in a multi row coil. It is conceivable that in a deep coil air dehumidification would take place in the last few rows of the coil, and that the average coil condition would not reflect this. For these cases, however, the dehumidification rate would be very small and therefore also the error made in considering an average row would be small.

4.2 The Finite Element Model

4.2.1 Assumptions of the Finite Element Model

The finite element model considers the average row of the coil in a number of geometrically fixed elements. The partition of a row into elements is shown in Figure 4.2. Each element is analyzed separately, and the total heat transfered is obtained by summing up the heat transfer in each element. The finite element model is subject to the following assumptions:

1. The state of the refrigerant in one element is either saturated or superheated.
2. The refrigerant temperature in one element can be represented by the inlet refrigerant temperature to that element. In Figure 4.3 of element j, this is $T_r(j)$. The elements are assumed small enough that this is a good assumption also in the case of superheated refrigerant. This allows use of equation (4.12) to determine the log-mean temperature difference in any element along the length of the average row.
3. The air mass flux, $(lb_m/(hr \cdot ft^2))$, along the length of the coil is constant. This assumption is only relevant for the frosted coil and is discussed further below.
4. The frosting process is quasi-steady, where parameters calculated at time t can be used for calculations at time $t+\Delta t$.
5. Frost and condensate can not form on the coil simultaneously. The coil is either all dry, all wet, all frosted, partially dry and partially frosted, or partially dry and partially wet.

4.2.2 Heat Transfer Analysis of an Element

The inlet refrigerant and air conditions have to be specified for a simulation. The air side inlet is determined by the ambient dry bulb temperature and relative humidity, and the dry air mass flow rate. The dry air mass flow rate can be determined by user specification of either: air mass flow, pressure drop across coil, or face velocity. Alternatively an air side pressure drop correlation together with a fan characteristic could be used to determine the volumetric flow rate of air. The advantage of specifying the flow is that variations with flow rates in coil performance can be studied readily, and the difficulty of finding a pressure drop correlation that is generally applicable can be avoided. In the modelling of the frosted coil the performance deterioration is mainly due to decreased air mass flow due to the rough frost surface and the frost flow obstruction. A fan characteristic together with a pressure drop correlation is therefore required to model the frosting process correctly. The air side pressure drop correlation used was discussed in section 3.8. A fan characteristic was obtained from meassurements taken using a pressure differential transducer installed on one of the coils used in the verification of these models. This fan characteristic together with the frosted coil friction factor equation were then used to obtain the correct flow rate as frost accumulated on the coil surface. The refrigerant inlet to an element is specified by: the saturation pressure or the saturation temperature, and the inlet enthalpy or the inlet quality. The refrigerant flow rate is determined by specifying the refrigerant mass flow rate, or the evaporator outlet condition, which takes the form of specified degree of superheat or specified outlet quality.

An iterative procedure is required to determine the heat transfer rate to an element of the average row. The first step in determining the heat transfer is to assume an average temperature. Initially the coil is assumed dry, and the parameter C, equation (4.4) is set equal to zero. Using the equations for the heat transfer coefficients given in Chapter 3 and equation (4.11) together with the assumed average temperature, the heat transfer to an element is determined. Once the heat transfer rate is determined the coil mean surface temperature can be estimated. The coil mean surface temperature in element j is given by:

$$T_{s,m} = T_{av} - q \cdot Res_{a,s,m} \tag{4.13}$$

where,

$$Res_{a,s,m} = \frac{1}{h_{tot} \cdot A_o \cdot n} \tag{4.14}$$

If the coil mean surface temperature is below the dew point of the air stream the parameter C must be determined, and the heat transfered to the element calculated again. Calculation of C requires the specific humidity at the average row. The method used in obtaining this is explained in the following paragraph.

When the air is dehumidified a process line on the psycrometric chart describing the states of the air stream as it passes through the coil is required in order to determine the leaving condition of the air stream. A straight line between temperature and humidity corresponding to incoming air and the coil surface temperature at saturation is assumed. An arbitrary process line is illustrated in Figure 4.4. The slope of this line is sensitive to the state of the inlet air but insensitive to the mean surface temperature. The slope of the process line is therefore a very good estimate of the parameter C in equation (4.4). Hence:

$$C = \frac{W_{av} - W_{s,m}}{T_{av} - T_{s,m}} \tag{4.15}$$

The calculation of the parameter C, the slope of the process line, is done independently in each element j. Notation for element j is omitted and it is understood that all heat transfer calculations are carried out independently in each element. As is shown in Figure 4.4 the specific humidity at the average condition is taken as the humidity corresponding to T_{av} on the process line. From an estimate of the average air temperature and the mean coil surface temperature the value of C can be determined, and a new heat transfer rate is calculated.

Once convergence on the heat transfered, and coil mean surface temperature, in an element for an assumed average temperature has been reached, the refrigerant and air leaving enthalpies can be calculated. The enthalpy of the air leaving an element of the average row is:

$$i_{a2} = i_{a1} - q/m_a \qquad (4.16)$$

and the enthalpy of the refrigerant at the end of an element is:

$$i_{r2} = i_{r1} + q/(n \cdot m_r) \qquad (4.17)$$

where q, and m_a are the heat transfer in one element, and the air mass flow across one element, respectively. The refrigerant mass flow, m_r is the mass flow in one circuit (one row). From the determined air enthalpy and the calculated slope of the process line the temperature and specific humidity of the air leaving the element under consideration can be determined using a bisection method. From Figure 4.4 it can be seen that the intersection of the process line with the constant enthalpy line, i_{a2}, determines the outlet condition of the air. From the calculated value of T_{a2}, and w_{a2} a new estimate of the average air temperature can be determined, using equation (4.12), and the new value is compared to the one used in the previous calculation. If they are equal to within specified tolerance the next element can be analyzed. If they are not sufficiently close, the new calculated value of the average temperature is used and the whole process described is repeated.

When all elements along the length of the coil have been analyzed the outlet condition of the refrigerant is compared to that specified, unless refrigerant mass flow was given in which case the simulation is completed. If the calculated outlet condition compare favourably with the specified condition the dehumidification rates, if any, and the frosting conditions, if the coil mean surface temperature is below 32°F, are determined.

If, on the other hand, the calculated do not agree with the specified refrigerant condition a new refrigerant flow rate is assumed. Figure 4.5 illustrates the variation of degree of superheat with refrigerant flow rate for a simulation using the finite element model. To obtain the flow rate corresponding to the desired outlet condition, one flow rate underestimating the outlet condition, and one overestimating it are obtained. Linear interpolation is then used to zero in on the correct flow rate. In the case of specified degree of superheat a linear interpolation between mass flow rate and degree of superheat is possible; however, when the outlet quality is specified the iterations must be performed on refrigerant enthalpy rather than refrigerant quality, since for all mass flows resulting in superheated refrigerant the quality is one. The enthalpy corresponding to the specified outlet quality at the inlet pressure is initially assumed. When convergence on the enthalpy value is obtained, the enthalpy value corresponding to the specified outlet quality and the calculated refrigerant pressure at the outlet is calculated. This enthalpy is compared to the previous value. If they agree to within acceptable tolerance the desired refrigerant mass flow is at hand, if they do not agree iterations about the new enthalpy are required. The computing time for only one iteration using the finite element model can be rather high. Iterating on the outlet condition can make the procedure prohibitively expensive. In addition to this the discrete nature of the finite element model sometimes cause convergence problems, which can be seen in Figure 4.5 ,where the average coil row was analyzed in 25 elements, by the non linear distribution of the performance points. These problems can be avoided by accepting larger tolerances or introducing smaller elements. The finite elmement model can be as exact as one can afford, (of course, limited by the error in heat transfer correlations) by decreasing the size of the elements; however, since heat exchanger analysis usually involves numerous simulations the finite element model often becomes impractical. This was the reason for developing the three region model, which contain some very reasonable assumptions, and is considerably faster, with no convergence problems. This model is explained in section 4.3 below. Once all elements have been analyzed and the desired outlet condition obtained, the condensation and frosting conditions are determined.

The rate of dehumidifiction in an element is determined from:

$$m_v = m_a \cdot (w_{a1} - w_{a2}) \tag{4.18}$$

If the mean coil surface temperature is above 32°F water will condense on the coil surface, and run of the coil due to the gravitational force. This is all the analysis required for the wet coil surface. At this point results are available. When the coil temperature is below 32°F, and the air is dehumidified, the desublimation process takes place and frost will accumulate on the coil. The frost accumulation is considered independently in each element. Using the frost property correlations presented in Chapter 3, the thermal resistance due to the frost can be determined in each element. This resistance is then used in the heat transfer equation (4.11) in the average row analysis at the next time step, t+Δt. The frost accumulation, and the frost thickness on the coil are obtained by summing up the frost accumulated in each element, and taking the average frost thickness based on all elements, respectively. Using the equations of Malhammar [1986], equations (3.35), (3.36), (3.58), and (3.59), the pressure drop across the coil can therefore be determined. The calculated pressure drop together with a fan characteristic then determines the air volumetric flow rate. From the specified inlet temperature the inlet air density is determined which allows calculation of the dry air mass flow rate. As previously mentioned the program allows the user to specify any one of initial air mass flow, initial air side pressure drop, or initial face velocity. The smaller of the mass flow corresponding to the user supplied air pressure drop of the non-frosted coil,(or the user specified mass flow or face velocity) and that obtained using Malhammar's equations is then used as the air mass flow rate at the next time step, t+Δt. This is not strictly correct since an air side pressure drop formula predicting a pressure drop for the smooth surface should be used. No correlations located in the literature came close to the relatively high pressure drop values observed for the coil used in the verification of the frosted coil model. As a result the approach described above was taken. From the calculated mass flow and the free flow through area (obstruction due to frost thickness taken into account) the air mass flux is determined. This mass flux is then used in the next iteration at time t+Δt, and is assumed constant along the length of the average row. As a result the less obstructed elements of the coil, in transition region and the superheated region, will be subject to a larger proportion of the total air mass flow. Unless the time simulated is equal to that specified, a time step, Δt, is taken and the calculations are repeated.

The output of the finite element model is rather extensive. The conditions at each element together with calculated average results are printed out. A sample output file is given in appendix 1. An outline of the iterative procedure used in the computer implementation of the finite element model is given in appendix 2, and the program listing is in appendix 3.

4.3 The Three Region Model

4.3.1 The Assumptions of the Three Region Model

This model divides the average row representative of the coil, into three different regions, each region depending on the refrigerant state. Figure 4.6 illustrates the two phase region (specified inlet quality to 0.75), the transition region (0.75 to 1.0), and the superheated region. The two-phase and transition regions are both obviously saturated. The reasons for choosing the transition region in the refrigerant quality range 0.75 to 1.0 was explained in section 3.5, and it was pointed out that the extent of the transition region is not always clear. In this model the heat transfer available in a region can be calculated from the refrigerant inlet and outlet qualities and the refrigerant mass flow rate. In the three region model the air side leaving condition and the area required for the available heat transfer in a particular region are the unknowns, as opposed to the finite element model where the air side and the refrigerant side leaving conditions are the unknowns and the area available for heat transfer is the known. The assumptions made in the three region model are:

1. The pressure drop on the refrigerant side can be neglected. As a result the refrigerant temperature in the two-phase and transition regions will be constant and equal to the saturation temperature.
2. Constant air mass flux. As was the case for the finite element model this becomes relevant only in case of the frosted coil.
3. The frosting process is quasi steady, and the time step is small enough that parameters calculated at time t can be used at time $t+\Delta t$.
4. Frosting and condensation can not occur simultaneously on the coil; however, the amount of air dehumidification is determined independently in each region.
5. The superheated region is dry. As heat is transferred to the refrigerant in the superheated state the temperature of the refrigerant increases. The assumption of a dry superheated

region allows use of the ϵ-NTU method for heat exchangers in cross flow to be used in the analysis of the average row in the superheated region.

The constant refrigerant pressure assumption was found to be very reasonable from a heat transfer point of view. Results from simulations using the finite element program with and without pressure drop (see Chapter 5) indicated that the errors associated with neglecting the pressure drop term, typically a fraction of one percent difference in heat and dehumidification rates for mass and heat density fluxes of interest for heating and cooling applications, were much smaller than those associated with uncertainties in heat transfer and pressure drop correlations. The calculated pressure drops were low and were found to be of the same order as the experimental results of Andersson [1966]. The heat transfer coefficient for the superheated refrigerant is an order of magnitude lower than the two-phase value, and the superheated refrigerant temperature increases quickly as heat is transferred to it. As a result the mean surface temperature of the coil in the refrigerant superheated region is considerably higher than in the saturated regions. Dehumidification in the superheated region will therefore only occur under condition of very moist air. Since the surface temperature is relatively high, the refrigerant side heat transfer coefficient low, and the region of superheat typically small (superheat is only protection for the compressor and is an ineffective way of evaporator use) the error associated with assuming a dry superheated region is small even in the case of moist air (φ > 90%) entering the heat exchanger. In most cases this assumption incurs no error.

4.3.2 Heat Transfer Analysis of the Regions

The procedure described for the finite element model largely carries over to the heat transfer analysis in the saturated regions (two-phase and transition) of the three region model. The analysis of the superheated region differs completely from the finite element model and is discussed separately further below. The analysis of the two-phase and the transition regions are identical, except for the value of the refrigerant side heat transfer coefficient. In the three region model the refrigerant side heat transfer coefficient in the transition region is calculated at the mean refrigerant quality in the region, which for the case illustrated in Figure 3.5 would be at a quality of 0.875. In the saturated regions the heat transfer available is calculated from:

$$q_{avl} = m_r \cdot i_{r,fg} \cdot (x_{out} - x_{in}) \tag{4.19}$$

The fraction of the coil required for this heat transfer rate, as determined from the heat transfer equations, is then assumed. Assuming an average air temperature and setting the parameter C initially to zero, dry coil, the method outlined in section 4.2.2, for the finite element model, is used to find the heat transfer as predicted from the heat transfer equations of Chapter 3, and equation (4.11). The heat transfer rate calculated from the heat transfer equations and the assumed area of the coil is then compared to the heat transfer rate available in the region under consideration, calculated using equation (4.19). If they agree to within acceptable tolerance the fraction of the coil in the region under consideration, for given average and surface temperatures, is at hand. If they do not agree, a new value for the fraction of the coil in the region considered is assumed. One fraction value overestimating the heat transfer and one underestimating it, as compared to that available, are obtained, after which linear interpolation can be used to quickly converge on the correct value for the coil fraction in the refrigerant region under consideration. When the fraction of the coil required in the region considered, for a given average temperature and surface temperature, is obtained, a new surface temperature is calculated. If the old and the new surface temperature are not equal to within tolerance the new value is used and the area fraction required for the available heat transfer is determined again. If the calculated surface temperature compares with the previous value the leaving air and refrigerant enthalpies are determined. The leaving state of the air is then obtained as outlined above, for the finite element model. The calculated T_{a2} is used to determine a new value for the average air temperature using equation (4.12). Convergence on the average temperature determines if analysis of a region is complete or not. If the average air temperature value from the previous iteration does not agree with the calculated value, a weighted average of the new and the old average temperature is calculated and the whole procedure outlined is repeated using the weighted average temperature. When convergence on the average temperature is obtained the air dehumidification rate and frost properties are calculated. If the fraction of the coil required to realize the latent heat capacity of the refrigerant in the region under consideration is greater than one, the fraction of the coil that is left for heat transfer in the region under consideration is determined (this would be 1 in the case of the two-phase region and 1 -fraction in two phase region, in the case of the transition region), and the heat transfer rate in the remaining fraction of the coil is calculated. In this case the situation is identical to the finite element model where the area is known, and air and refrigerant leaving conditions are to be determined in the region, or element, analyzed. When convergence on the average air temperature has been achieved, and the fraction of the coil used is less than or equal to one, the analysis of the region is complete. If the analysis of the two-phase region was completed the transition region is analyzed using the procedure outlined above. If the transition region analysis was completed then the superheated refrigerant region is analyzed next, as described by the following procedure.

When the fraction of the coil required for the evaporation of the refrigerant has been determined, and is less than one, the superheated region is analyzed. The ε-NTU method suggested by Kays and London [1984] for cross flow heat exchangers is adopted. Because of the irregular circuitry of the refrigerant tubes, the net effect is mixing of the air stream. The equations for both fluids mixed are therefore used. The coil heat transfer effectiveness, ε is defined as:

$$\varepsilon = \frac{q}{q_{max}} = \frac{C_{air} \cdot (T_{ai} - T_{a2})}{C_{min} \cdot (T_{ai} - T_{ri})}$$

$$= \frac{C_{ref} \cdot (T_{r2} - T_{ri})}{C_{min} \cdot (T_{ai} - T_{ri})} \qquad (4.20)$$

where,

$$C_{ref} = m_r \cdot C_{p,r} \qquad (4.21.1)$$

$$C_{air} = m_a \cdot C_{p,a} \qquad (4.21.2)$$

and,

$$C_{min} = \min (C_{air}, C_{ref}) \qquad (4.22.1)$$

$$C_{max} = \max (C_{air}, C_{ref}) \qquad (4.22.2)$$

let,

$$r = \frac{C_{min}}{C_{max}} \qquad (4.23)$$

The number of transfer units, NTU, is given by:

$$NTU = \frac{UA}{C_{min}} \qquad (4.24)$$

where,

$$UA = 1 / \sum_i Res_i \qquad (4.24.1)$$

and the thermal resistance terms are given by equations (4.5) to (4.10) and the heat transfer correlations of Chapter 3.

The heat transfer effectiveness is then given by:

$$\varepsilon = \frac{NTU}{\left[\frac{NTU}{1 - EXP\ (-NTU)} + \frac{r \cdot NTU}{1 - EXP\ (-r \cdot NTU)} - 1\right]} \tag{4.25}$$

The heat transferred to the superheated refrigerant is therefore:

$$q = \varepsilon \cdot C_{min} \cdot (T_{a1} - T_{r1}) \tag{4.26}$$

and the air leaving temperature in the superheated region is:

$$T_{a2} = T_{a1} - \varepsilon \cdot r \cdot (T_{a1} - T_{r1}) \tag{4.27}$$

and the degree of superheat is given by:

$$\Delta T_{sup} = \varepsilon \cdot r \cdot (T_{a1} - T_{r1}) \tag{4.28}$$

and the refrigerant temperature leaving the coil:

$$T_{r2} = \Delta T_{sup} + T_{r1} \tag{4.29}$$

Since the superheated region is dry, $w_{a2} = w_{a1}$. At this point all three regions along the average row of the coil is analyzed and the evaporator leaving condition is available.

The procedure for converging on the refrigerant mass flow rate corresponding to a specified outlet condition is identical to the method described in the finite element model. The convergence on specified outlet quality is simpler for the three region model since no pressure drop is considered, and as a result the enthalpy corresponding to the enthalpy at the inlet saturation pressure and specified outlet quality is the correct enthalpy.

For the dehumidifying case the rate of condensation or frosting is determined in the two-phase and the transition regions. In the analysis of the frosted coil for the three region model the frost accumulated in the saturated regions is calculated. The refrigerant regions are not stationary, whereas the frost accumulated obviously does not move with the refrigerant regions. Depending on the relative movement of the transition and the two-phase regions, frost that was in a particular region at time t, will be in the other region at time t+Δt. For example if the two-phase region is retarded, frost previously in the two-phase region will become part of the transition region. The three region program outline in appendix 2 explains this point further. Once the accumulated frost in each region is determined, the three regions of the coil are treated in exactly the same way as the elements in the finite element model. The frost thermal resistance, and the pressure drop and the associated air mass flow is determined and used at the next time iteration. The air mass flux determined from the air mass flow and the free flow through area at time t is used to determine the mass flows in the different regions at time t+Δt assuming that the mass flux stays constant along the length of the coil.

Since the three region model is computationally faster than the finite element model, and the assumptions made were found to be very reasonable, almost all simulations were made using the three region model. Additional parameters, not directly indicative of coil performance, for use in the derivation of the parametric model where therefore calculated in the three region computer program. These parameters are coil characteristics and coil effectiveness. The three region computer program also allows for a schedule of inlet conditions to be specified. This was required for the verification of the frosted coil model since the coils used were subject to outside air conditions which, of course, vary throughout the day. This is elaborated on in Chapter 5, discussing the model verification. A sample output file of the three region model program is given in appendix 10.

4.4 The Parametric Model

A short and numerically fast evaporator model was desired as part of a larger heat pump simulation program. The model derived for this purpose is a purely empirical model. The data required for the numerical model is conveniently obtained from the finite element or, preferably due to computational efficiency, from the three region model. The required data could also be obtained from experimentation.

The performance of a dehumidifying coil can be described by two parameters; the coil characteristic and the coil enthalpy effectiveness, similar to ASHRAE Equipment [1983]. These two parameters are defined here as:

$$CC = \frac{T_{adp} - T_{ri}}{i_{ai} - i_{adp}} \tag{4.30}$$

$$\varepsilon_i = \frac{i_{ai} - i_{ao}}{i_{ai} - i_{adp}} \tag{4.31}$$

For the dry coil the coil characteristic becomes superflous and the coil performance can be described by the coil temperature effectiveness:

$$\varepsilon_T = \frac{T_{ai} - T_{ao}}{T_{ai} - T_{ri}} \tag{4.32}$$

In ASHRAE Equipment [1983] the coil characteristic and the coil effectiveness definitions include the average air condition in place of the inlet air condition as written in equations (4.30), (4.31), and (4.32). The coil performance was found to be determined just as well by the coil parameters defined above as it was using average air conditions. Since the calulations using the coil parameters defined in terms of the inlet air condition simplifies the determination of the air leaving condition, they were adopted.

In ASHRAE Equipment [1983] it is shown that the coil characteristic is a function of the total air side thermal resistance (the effects of dehumidification included), the refrigerant side thermal resistance, and the thermal resistance of the wall. It can therefore be expected that the coil characteristic is a function of any of the parameters affecting the above mentioned resistances. The parameters affecting the coil characteristic is assumed to affect the coil effectiveness parameters as well.

The apparatus dew point has been introduced in the calculations of the coil parameters. The apparatus dew point is the saturated coil condition on the straight line defined by the air inlet condition and the air leaving condition. The apparatus dew point is therefore identical to the mean surface coil temperature at the average conditions as calculated in equation (4.13), except in the case when the average wall temperature is higher than the leaving temperature. This is possible in this modelling approach, for deep coils. For deep coils the enthalpy calculated from equation (4.16) can fall below the enthalpy corresponding to the saturated enthalpy at the coil mean surface temperature in which case the leaving condition is assumed saturated. The apparatus dew point is then equal to the air leaving condition and the coil effectiveness in equation (4.31) is equal to one.

For a constant, given refrigerant outlet condition the coil characteristic and the coil effectiveness were found to be well represented by the following general equations:

$$\Delta CC = c_1 \Delta T_{a,r} + c_2 \Delta T_{a,r}^2 + c_3 \Delta\varphi + c_4 \Delta\varphi^2 + c_5 \Delta V_{fc} + c_6 \Delta V_{fc}^2 + c_7 \Delta x_{in} + c_8 \Delta x_{in}^2 \tag{4.33}$$

$$\Delta e_i = e_1 \Delta T_{a,r} + e_2 \Delta T_{a,r}^2 + e_3 \Delta\varphi + e_4 \Delta\varphi^2 + e_5 \Delta V_{fc} + e_6 \Delta V_{fc}^2 + e_7 \Delta x_{in} + e_8 \Delta x_{in}^2 \tag{4.34}$$

$$\Delta e_r = b_1 \Delta T_{a,r} + b_2 \Delta T_{a,r}^2 + b_3 \Delta V_{fc} + b_4 \Delta V_{fc}^2 + b_5 \Delta x_{in} + b_6 \Delta x_{in}^2 \tag{4.35}$$

where,

$$\Delta CC = CC - CC_{ref} \tag{4.36.1}$$

$$\Delta \varepsilon_{i} = \varepsilon_{i} - (\varepsilon_{i})_{ref} \tag{4.36.2}$$

$$\Delta \varepsilon_{T} = \varepsilon_{T} - (\varepsilon_{T})_{ref} \tag{4.36.3}$$

$$\Delta T_{a,r} = (T_a - T_r) - (T_a - T_r)_{ref} \tag{4.36.4}$$

$$\Delta \varphi = \varphi - \varphi_{ref} \tag{4.36.5}$$

$$\Delta V_{fc} = V_{fc} - (V_{fc})_{ref} \tag{4.36.6}$$

$$\Delta x_{in} = x_{in} - (x_{in})_{ref} \tag{4.36.7}$$

In many cases a linear variation of the coil characteristic and effectiveness with respect to a particular influence parameter was observed, allowing the quadratic term to be dropped. Equations (4.33) and (4.34) were used for the wet and frosted coils and equation (4.35) was used for the dry coil.

The three region computer program was used to calculate the coil caracteristic and effectiveness for the reference state, and the required simulations to obtain the constants c_i, e_i, and b_i (i = 1 to 8). By varying one of the parameters in equations (4.36) at a time the variation in the coil characteristic and the coil effectiveness with that particular parameter could be determined. A number of points for each parameter in equations (4.36) were plotted against the change in the coil parameter relative to the reference state. A curve fit passing through the origin was then obtained for each parameter. A different reference state for each coil condition, dry, wet, or frosted, was necessary. As a result five equations for one coil will be required, and the user will have to make the desicion which equation is suitable under prevailing conditions.

Having obtained the appropriate constant values in equations (4.33), (4.34), and (4.35) the coil characteristic and coil effectiveness at conditions other than the reference state, but at the same coil condition, can be calculated from these equations. From calculated values of coil effectiveness and coil characteristic the state of the leaving air and refrigerant are determined by solving equations (4.30) and (4.31) simultaneously through iterative means for the wet and frosted coil, and by solving equation (4.33), a one step procedure, in the case of dry coil surface. When the leaving air condition is determined the heat transfer rate is calculated from:

$$q = m_a \cdot (i_{a1} - i_{a2}) \qquad (4.37)$$

and the dehumidification rate is:

$$m_v = m_a \cdot (w_{a1} - w_{a2}) \qquad (4.38)$$

For the wet and the dry coil this is essentially all the analysis required. The frosted coil does, however, present some unique problems.

The performance deterioration of the frosted coil is a result of added thermal resistance due to the frost layer, and reduced air flow due to obstructed flow through area and the rough frost surface. From the simulations, and the results of other investigators (for example Gates et al. [1967]), the main source of performance deterioration stems from the reduced air flow due to the increased pressure drop resulting from the frosted surface. The thermal resistance of the frost layer would typically be a few percent of the total thermal resistance, except in cases of very low density frost when the frost layer thermal resistance is of more relative importance;however, in the case of low density frost, also the pressure drop would increase rapidly due to flow obstruction. Since the parametric model does not contain any heat transfer equations the effect of the frost thermal resistance can not be conveniently included. The equations of Malhammar [1986] were used, however, to determine the frost density which together with the coil geometry and the accumulated frost was used to calculate the frost layer thickness, which allows determination of the local air velocity as defined by equation (3.35). From the friction factor given by equation (3.58), and the local air velocity the pressure drop is determined from equation (3.59). Using the fan characteristic the air mass flow corresponding to the pressure drop is determined. The same criteria as for the three region and finite element model is then used to determine the flow rate at the next time level. In other words if the calculated air mass flow rate is lower than the initial user specified value it is used at the next time level. If the air mass flow is reduced the face valocity will decrease and, as a result, the coil characteristic and the coil effectiveness need to be calculated again. If there is no change in the air mass flow rate the coil characteristic and the coil effectiveness are assumed to stay constant.

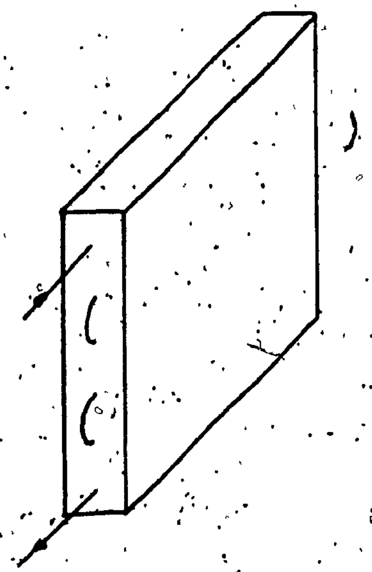


FIGURE 4.1 a

One Row Coil

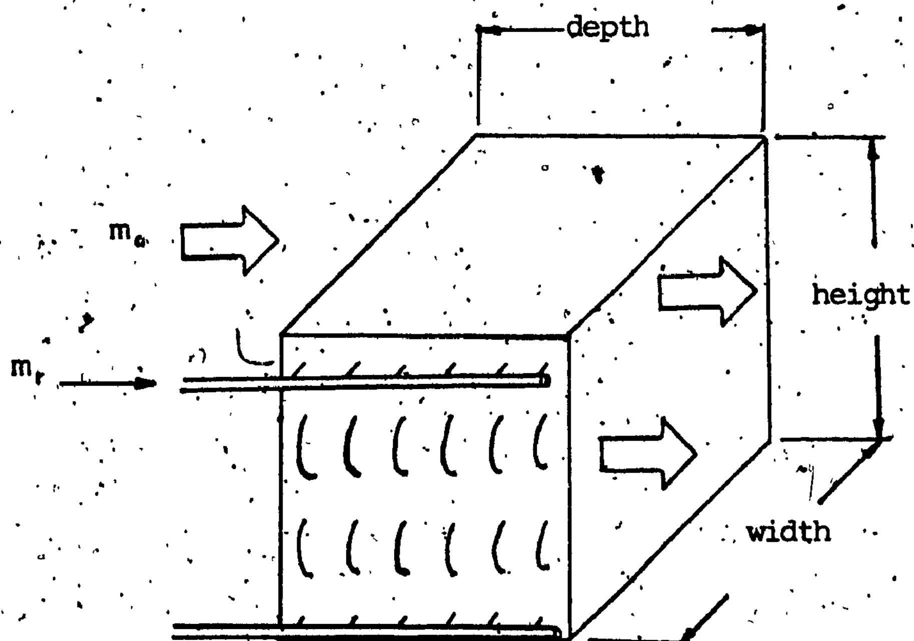


FIGURE 4.1 b

Multi Row Coil

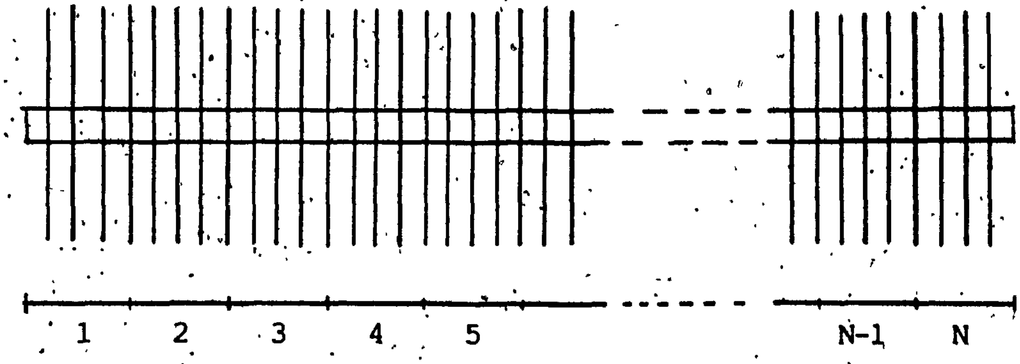


FIGURE 4.2

The Average Row Element Partition

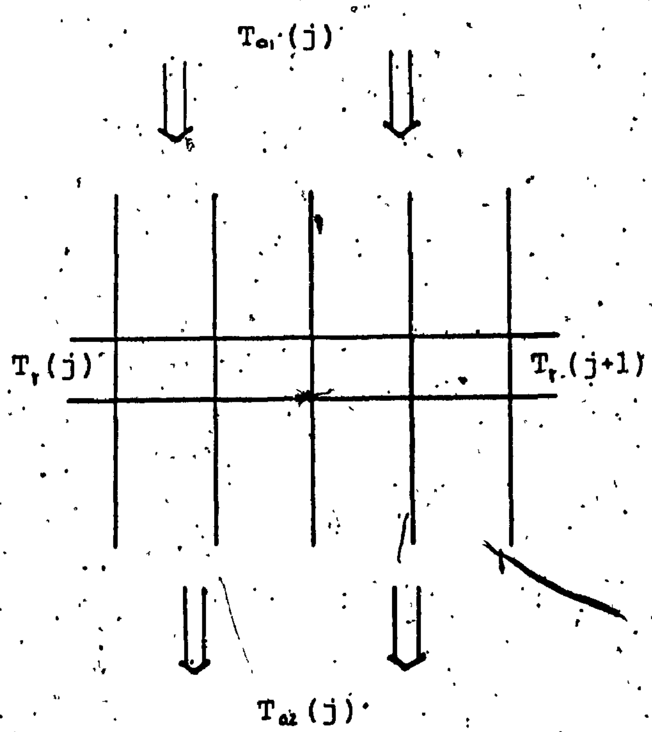


FIGURE 4.3

The j-th Element

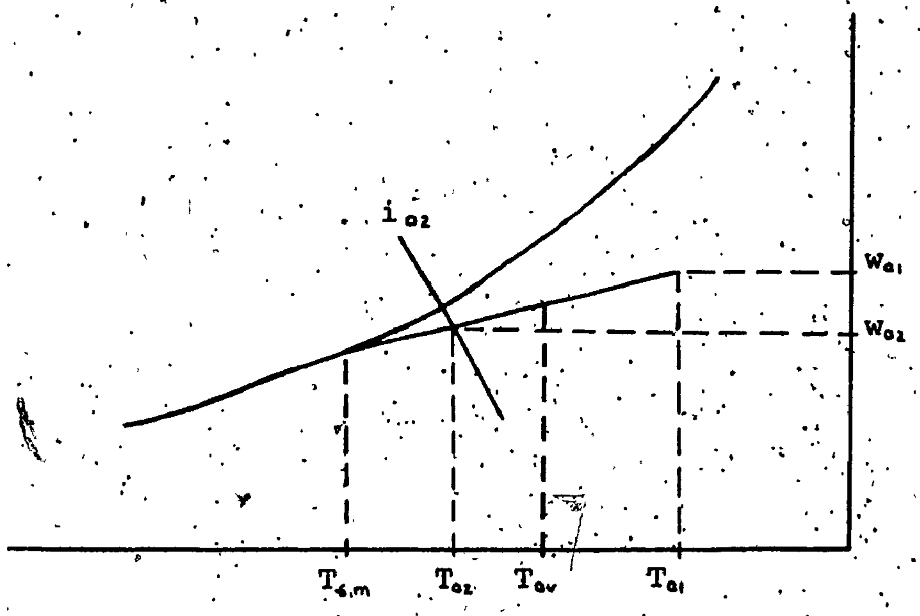


FIGURE 4.4

Air Process Line on Psycrometric Chart

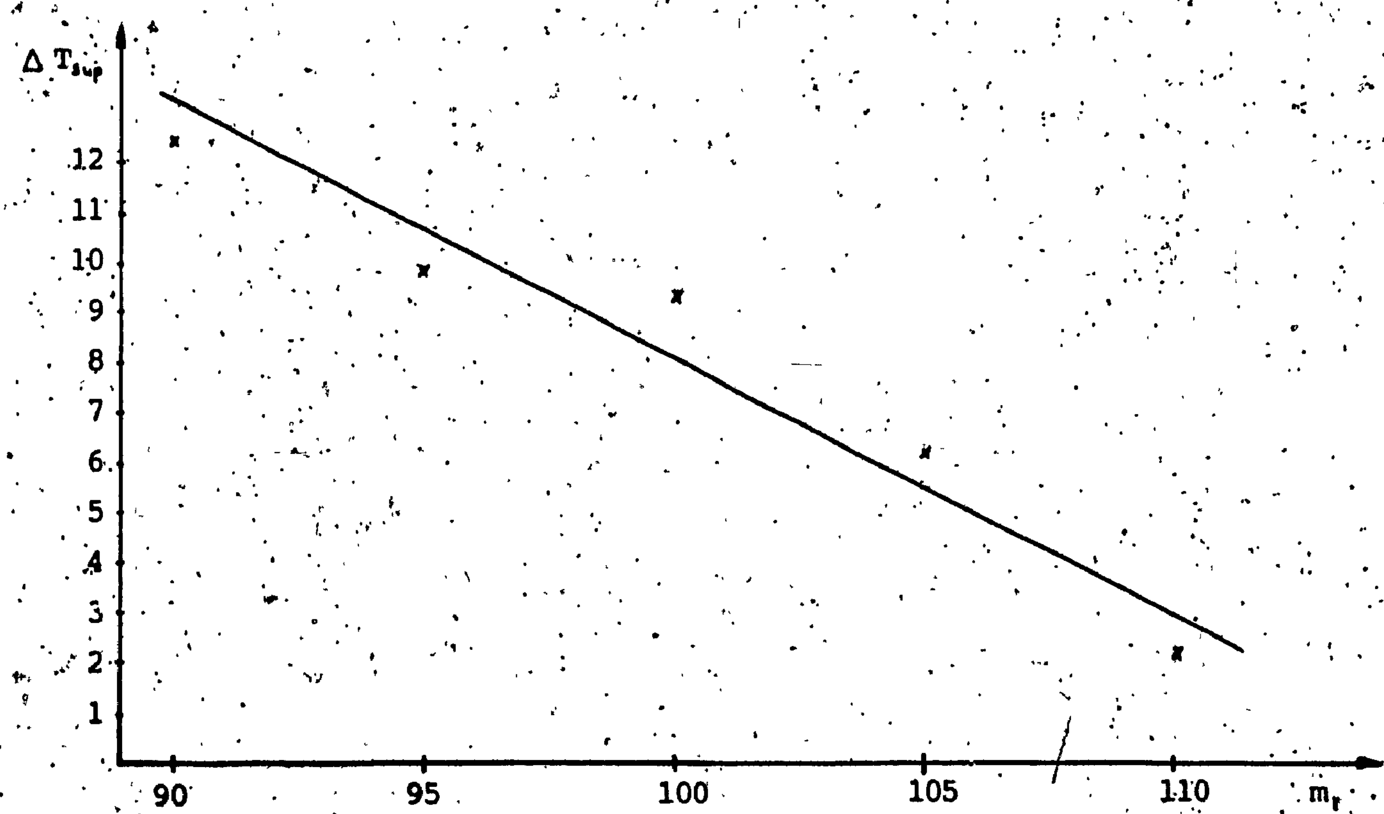


FIGURE 4.5

Finite Element Model Predicted Superheat

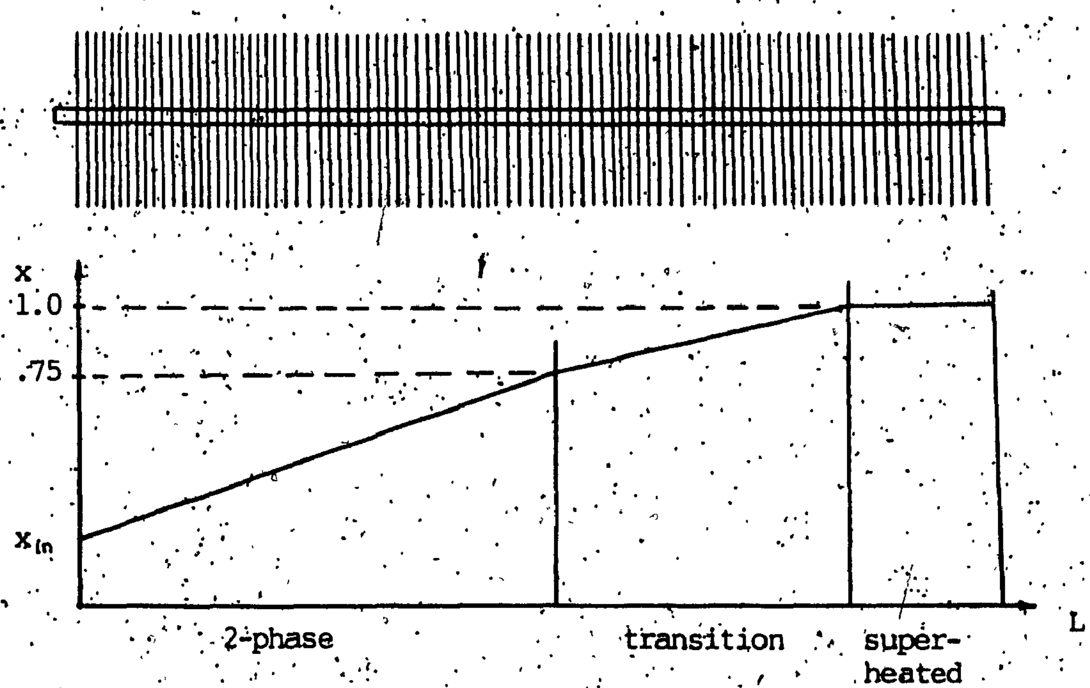


FIGURE 4.6

Three Region Partition

CHAPTER 5

VERIFICATION OF MODELS

5.1 Evaporator Coils Used in the Verification.

Experiments on two evaporators were used to verify the evaporator models. One coil had six circuits (rows) arranged such that the air flowed through the coil in a serpentine manner, as shown in Figure 5.1. The six row coil has the following dimensions:

A_{fc} = 4.0 ft^2

L = 32.0 ft per circuit (row)

fpi = 8 fins per inch

y = 0.006 inch

d_1 = 1.50 inch

d_2 = 1.50 inch

D_i = 0.500 inch

D_o = 0.506 inch

A second coil was installed in late February 1988. This was a simple one row coil, with the following dimensions:

A_{fc} = 4.725 ft^2

L = 37.8 ft

fpi = 8 fins per inch

y = 0.006 inch

d_1 = 1.50 inch

d_2 = 1.50 inch

D_i = 0.500 inch

D_o = 0.506 inch

The six row coil has been in operation much longer than the one row coil and much more data is therefore available, and used, on this coil. It was also found that the one row coil was not very suitable for wet and frosted coil tests. Since the one row coil is shallow, the air stream temperature drop is small, and only in the case of very moist air will frost or condensate deposit on the coil surface. An error in the air condition (temperature and relative humidity) reading would affect the one row coil simulated results much more than it would the six row coil's. The evaporators were located outdoors, and were therefore subject to daily temperature and humidity variations. This was not a problem for the dry and wet coil model verification; however, for the frosted coil verification the lack of control resulted in some problems. This is discussed further in the presentation of the results below.

5.2 Experimental errors

A data acquisition system was used to scan the meassurement points once every 18 seconds, averaging results every 30 minutes. The time averaged results were then used to obtain the desired quantities. The heat transfer rate was determined from meassurements taken on the refrigerant side. The system meassurements were analyzed by a computer and heat transfer rates were obtained using pure refrigerant property subroutines. The air dehumidification rate was based on readings of air temperature, and relative humidity probes. Although great care was excercised in the analysis of the results, there are always errors inherent in an experimental set up.

5.2.1 Errors in instrumentation

Experimentation errors occur on the refrigerant side, affecting the experimentally determined heat transfer rate and evaporator outlet condition, and on the air side, affecting experimentally determined air dehumidification rates. Reading errors in the leaving and entering air conditions affect the experimental, and model prediction of dehumidification rates, which can cause experimental results and model predictions to deviate extensively. The frosted coil is a transient problem and errors accumulate with time, making meassurement errors relatively more significant in the verification of the frosted coil as compared to the dry and wet coil. A relatively small reading error in the relative humidity can have a substantial effect on the calculated dehumidification rate. Malhammar [1986] noted from his work that a 1% reading error in the relative humidity of the entering air stream results in a 9% error in the dehumidification rate, where the air dehumidification rate is determined from meassurements on entering and leaving relative humidities. Air temperature meassurement errors does not affect the experimental heat transfer prediction. It does, however, greatly affect the model predicted heat transfer rates, since heat transfer is here calculated with air to refrigerant temperature difference as the heat transfer driving force. As a result an error in the inlet air temperature, and/or relative humidity meassurement may result in model and experimental results deviating substantially.

The major source of error on the refrigerant side is the evaporator pressure meassurement and the determination of the evaporator leaving condition. Whereas the effect of these parameters on the experimentally determined heat transfer rate is small, the effect on the heat transfer rate predicted by the models can be very significant. A meassurement error in the evaporator pressure results in an error in the refrigerant saturation temperature which affects the predicted heat transfer in the same way as an error in the inlet air temperature. Incorrect determination of the leaving evaporator outlet condition significantly affects the model predictions. The experimentally determined outlet condition is an input to the models and is obtained by iterating on the refrigerant mass flow rate. The refrigerant mass flow rate has an effect on all modelled parameters.

5.2.2 Effects of Expansion Valve Hunting

The evaporator leaving condition was controlled by a thermostatic expansion valve. The evaporator coils were tested in heat pumps having an interchanger (suction line liquid subcooling heat exchanger). The point of control was either the evaporator outlet, to achieve a superheated condition at the evaporator outlet, or the interchanger outlet, in which case the evaporator outlet would be either wet or superheated. The expansion valve control is, however, by no means perfect. Using instantaneous data (scans approximately 6 seconds apart) a time plot of refrigerant degree of superheat at the evaporator outlet indicate severe hunting around the desired outlet condition. Figure 5.2 illustrates this qualitatively. The effects of expansion valve hunting is most significant at low degrees of superheat. In the analysis of the experimental data, time averaged temperature and pressure meassurements are used to determine the corresponding enthalpy. As can be seen from Figure 5.2 the time average would indicate a superheated leaving condition, whereas a mass average could result in a saturated leaving condition, depending on the degree of wetness. As a result, the experimentally determined time average does not necessarily reflect the, thermodynamically correct, mass average leaving condition. As mentioned, the evaporator outlet condition is an input for the models. An incorrect estimate of the mass average leaving condition would therefore be a source of discrepancy between model predictions, and experimental results.

5.2.3 Oil in the Refrigerant

The heat pump system used for testing of the six row coil contained an oil separator. Although an oil separator is used some oil will mix with the refrigerant and flow through the heat pump cycle. The heat pump system used for the testing of the one row coil did not have an oil separator. The properties of pure refrigerants are used in the analysis of the experimental data and in the models. Investigators in the field appear to agree that the heat transfer and physical properties of pure refrigerants are different from those of oil bearing refrigerants, and that the effect depends on the degree of miscibility of the oil in the refrigerant. The results of various investigations were discussed in section 2.2. In the work of Chaddock [1986] it was noted that the refrigerant side heat transfer coefficient for Refrigerants 12 and 22 (used in the verification of the models) was enhanced for a certain range of qualities, and reduced in the latter part of the saturated region. The degree of enhancement was found to be a function of amount of oil in the refrigerant (Schlager et al. [1988], Chaddock [1986]). The work of Hughes et al. [1982] indicate that thermodynamic properties of pure refrigerants are insufficiently accurate for oil concentrations greater than 1%. The exact amount of oil in the refrigerant is often impossible to determine. Hughes et al. [1982] present enthalpy pressure charts for Refrigerant-12 with oil concentrations: 0.1, 2.0, 5.0, 8.0, 10.0, and 15.0 percent. To incorporate these results in numerical modelling a set of equations have to be developed, and the remainiq physical properties included (specific volume, specific heat, entropy). Chaddock [1986] presents a correlation for the refrigerant heat transfer coefficient at low oil concentrations. It is clear that more work is required in this field to allow the effect of oil bearing refrigerant to be included in evaporator (and condenser) modelling. Due to unknown oil concentration and incomplete theories on the effect of oils, it is difficult to determine the effect on the experimental reults and the model predictions. It is, however, clear that it is a source of discrepancy between the two.

5.3 Model Verification

The models are verified by comparing three region model predictions with experimental results. This is accomplished by first establishing the agreement of finite element and three region model predictions. The finite elment model is most fundamental; it contains the least number of assumptions. The agreement of three region model predictions and experimental results will therefore verify both the finite element and the three region model. The parametric model is then verified against the three region model.

5.3.1 Three Region and Finite Element Model Predictions

Tables 5.1 and 5.2 compare the three region model and the finite element model predictions. Table 5.1 show five operating points, four wet, and one dry. In Table 5.2, two simulations for a frosted coil are given. The refrigerant flow rates, for the wet and dry coil conditions, are in the higher range of what was observed in the tests on the coils, as to make the refrigerant pressure drop more significant. It can be seen that the results of the three region model compare very well with those of the finite element model. The heat transfer rate as predicted by the three region model was observed to be within ±2% of the finite element predictions under the conditions encountered for the coils. Often the deviations were less than ±1%. The deviation in the dehumidification rate is greater (therefore also frost thickness and frost accumulation in Table 5.2), but the predictions are still very close. At low dehumidification rates the percentage errors can become very large. A comparison of the absolute values indicate, however, that the three region model predictions are very satisfactory. The largest error appear in the degree of superheat. This error can not be attributed to the omission of the refrigerant pressure drop in the three region model, alone. Table 5.11, showing predictions of the finite element model with and without refrigerant side pressure drop accounted for, indicate a small error in the degree of superheat resulting from the omission of the refrigerant side pressure drop. Two operating points are illustrated in Table 5.11. Point 1 has a refrigerant mass flow twice that of point 2, and both have a percentage error less than 2% in the degree of superheat. It was clear from the tests that the error in the degree of superheat is mainly due to the discrete nature of the finite element model as compared to the three region model which is continous. The refrigerant temperature increases quickly in the superheated region. An element can only have refrigerant in one state. As a result when the quality of the refrigerant reaches 1 in an element, the temperature of the refrigerant will only start to increase in the next element. As a result the leaving degree of superheat is underestimated in the case of the finite element model. This is more noticeable at low degrees of superheat. The methods for analyzing the heat flow in the superheated region are also different between the two models. The three region model uses the e-NTU method, whereas the finite element model uses the heat transfer equations as outlined previously in Chapter 4. As would be expected the agreement on the outlet condition is much better when outlet quality is specified (operating points 2, and 3).

5.3.2 Three Region Model Predictions Compared to Experimental Results

Three region model predictions are compared with experimental results on the six row coil for dry and wet surface conditions in Table 5.3, and for the one row coil in Table 5.4. The air mass flow rate used in the simulations of the six row coil was that corresponding to the air side pressure drop on the non-frosted coil surface. A differential pressure transducer was installed, and the air volumetric flow rate was correlated with the pressure drop. The non-frosted coil surface had a pressure drop of 0.567 in. w. g., corresponding to a flow of 1258 cfm. The air mass flow was then calculated from this flow rate and the air density corresponding to the inlet temperature. For the one row coil the air mass flow was obtained from meassuring the face velocity. The face velocity for the non-frosted coil was found to be 617.8 fpm, and was used for the simulations in Table 5.4.

For the six row coil simulations the predicted heat transfer rates were found to be within 15% of the experimentally observed value, with a few points deviating more than 15%. Many of the simulations were within 10% of the experiments, as is seen from Tables 5.3 and 5.4. The error in the dehumidification rate is larger, and the model prediction indicate a dry coil in some cases when experimental results suggests condensation (points 3, 7, 12, 13, and all of Table 5.4, discussed further below). This is to be expected, especially at low dehumidification rates, where percentage errors are large even for small absolute differences. It was also noted above that the experimentally determined dehumidification rate is extremely sensitive to the humidity readings. The leaving temperature prediction is generally lower than the meassured value. This can be attributed to the fact that the model appear to overestimate the heat transfer rate rather than underestimate for the six row coil. The predicted leaving humidity for the six row deep coil was generally higher than the meassured value. This was not the case for the one row coil, although the relative humidity meassurements are suspect on this coil. From Table 5.4 it is seen that most operating points are wet according to experimental readings. The three region model predictions, however, indicate a dry coil. The relative humidity readings have to be questioned. Both the air temperatures and the relative humidities are in many cases low and considering that the coil is only one row deep, dehumidification of the air stream is not to be expected. In fact, examination of the points on a psychrometric chart indicate that in many cases the evaporation temperature is greater than the dew point of the inlet air stream. Clearly under these conditions no dehumidification can take place. The error in the heat transfer error is smaller for the one row coil. This is to be expected due to the modelling approach taken in which an average row is analyzed for a multi-row coil.

As one would expect the errors are larger in the verification of the frosted coil model. Three simulations for the frosted six row coil are included in Tables 5.5, 5.6, and 5.7. These are termed simulation (A), (B), and (C), respectively. The air flow in the simulation was obtained from the calculated pressure drop using the results of Malhammar [1986], discussed in Chapter 3, and the flow rate versus pressure drop curve obtained from meassurements on the six row coil. The curve obtained was found to be well described by the straight line:

$$CFM = -534.03 \cdot \Delta P_{air} + 1560.7 \qquad (5.1)$$

in the region of interest, where CFM is in units of (ft^3/min) and ΔP_{air} is in units of (in. w.g.).

In all frosted coil simulations the inlet conditions used are the actual time varying ambient conditions, which are updated every hour of simulation. These time varying conditions were found to be a problem in the frost property determination. The frost density value is sensitive to the ratio of latent to sensible heat transport. In simulation (A), (Table 5.5) for example, the relative humidity value jumps from 80% to 70% in two hours of simulation. This has the effect of increasing frost density substantially, even though time averages are used in the equations, and decreasing the frost thickness as a result. A decrease in frost thickness obviously does not occur in reality (as long as surface temperatures are below 32°F), but is an undesired possibility in the numerical modelling. The increased density has the effect of increasing frost thermal conductivity, and increasing free flow area (if not in absolute terms, at least relative to what would have been the case if the relative humidity had been constant). The increasing error in the heat transfer rate with time observed in Table 5.5 can therefore partly be explained by an overestimate of the frost density in the numerical model, and therefore the thermal resistance due to the frost layer is larger than what is predicted numerically. The air flow rate in simulation (A) starts to decrease at ten hours of simulation (before this the volumetric flow rate is held constant for reasons discussed in both Chapters 3 and 4). The experimentally determined flow rate has during this time period decreased from 5626 to 5298 lb_m/hr. The model predicted flowrate starts to decrease rather rapidly compared to the experimental value from this point on. At 15 hours where the simulation is terminated due to coil frost up, the model predicted flow rate is actually less than the experimentally determined value. This can only be explained by the fact that frost is permeable and air passes through the frost which at this point has more or less 'blocked' the coil. The numerical model has no means of accounting for the permeability of the frost layer. As a result the air velocity becomes larger than would be the case if permeability was accounted for, which result in larger pressure drop and larger reduction in the air volumetric flow rate.

The trends observed in the different simulations are rather similar. In simulation (B), (Table 5.6) the air flow rate is modelled closer, but again the heat transfer rate error increases gradually from being underestimated by 15% at the outset to being overestimated by 21% after 10 hours of simulation. The dehumidification rate is underestimated for the first eight hours, which results in decreased frost accumulation, and increased frost density which have the effect of increasing free flow through area and decreasing frost thermal resistance. This has the effect of increased predicted heat transfer relative to the experimental values. The trends in simulation (C), (Table 5.7) are similar to those of simulations (A), and (B), and requires no further discussion.

It is clear that the transient frosted coil process simulation is very sensitive to variations in the inlet conditions. Particularly air humidity and temperature as well as refrigerant saturation temperature. A small discrepancy in the dehumidification rate can have significant effects on the results a number of hours later. In Table 5.12 the average inlet conditions (obtained from Tables 5.5, 5.6, and 5.7) for simulations (A), (B), and (C) are given. In Figures 5.3, 5.4, and 5.5 the time simulations of the frosted coil process for simulations (A); (B), and (C), respectively, using the average inlet condition, are illustrated graphically. These graphs illustrate much more clearly the effect of frost on the coil and the effect of the inlet parameters on the coil performance. As an example ,the relative humidity for simulation (B) is 88% as compared to 79% for simulation (A). After 13 hours the simulated frost layer thickness is observed to be approximately the same for the two simulations;however, the accumulated frost in simulation (A), Figure 5.3, is 41 lb_m as compared to simulation (B), Figure 5.4, where 34 lb_m of frost has accumulated on the coil. For simulation (A) and (B) where the coil is obstructed by frost rather quickly the constant air mass flow rate has the effect of increasing the local air velocity. This increases air side heat transfer coefficients and also dehumidification rates. The result is a slight increase in the heat transfer rate until the air mass flow rate starts decreasing due to the increased pressure drop.

5.3.3 Verification of the Parametric Model

The parametric model equations (4.33), (4.34) and (4.35) were verified for a six row coil using the three region model. The constants in the model eqations were obtained using the three region model as described in section 4.4. As mentioned the constants could also be determined using experimental results if sufficient data is available. Using equations (4.33), (4.34), and (4.35) assumes a constant outlet condition. In the verification, 10 °F superheat was used in all simulations. A different set of equations was determined necessary for each different coil condition. For the dry coil, only equation (4.35) is required. The reference state for the dry coil was selected as:

$$T_a = 50\ °F$$
$$\varphi = 0.35,$$
$$T_r = 24\ °F$$
$$V_{f/c} = 300\ fpm$$
$$x_{in} = 0.2$$

and the parametric equation (4.35) was determined as:

$$\Delta e_T = 0.01207 \cdot \Delta T_{a,r} - 4.592 \cdot 10^{-4} \cdot \Delta T_{a,r}^2 - 5.0 \cdot 10^{-4} \cdot \Delta V_{fc} + 0.12 \cdot \Delta x_{in} \qquad (5.2)$$

As can be seen from Table 5.8 this equation correlated the results of the three region model very well. With air temperature in the range 40 to 60 °F and refrigerant (saturation) temperatures in the range 17 to 33 °F the predicted heat transfer rate was correlated within ±5%. The inlet air humidity can take on any value as long as the air stream is not dehumidified. The predicted outlet temperatures were also very close between the two models as is illustrated in Table 5.8. The errors will be largest when the extremes of the range of the parameters considered in Table 5.8 are combined. Some combinations of these variables will undoubtedly lead to larger errors than those in Table 5.8. Some user judgement is required, as always when empirical type equations are used.

The verification of the parametric model for the wet coil is illustrated by ten points in Table 5.9. The reference state was selected as:

$$T_a = 50\ °F$$
$$T_r = 24\ °F$$
$$\varphi = 0.70$$
$$V_{fc} = 300\ \text{fpm}$$
$$x_{in} = 0.2$$

and the parametric equations (4.33), and (4.34) were determined as:

$$\Delta CC = -0.1332\cdot\Delta T_{a,r} + 0.001967\cdot\Delta T_{a,r}^2 - 0.580\cdot\Delta\varphi - 8.90\cdot\Delta\varphi^2 + 0.00847\Delta V_{fc} - 1.760\ \Delta x_{in} \qquad (5.3)$$

$$\Delta\varepsilon_i = 0.0194\cdot\Delta T_{a,r} + 0.720\cdot\Delta\varphi - 1.30\cdot10^{-4}\cdot\Delta V_{fc} + 0.03\cdot\Delta x_{in} \qquad (5.4)$$

$$\text{If } \varepsilon_i > 1.0\ ,\ \text{set } \varepsilon_i = 1.0 \qquad (5.4.1)$$

These equations do not correlate the three region, wet coil results as well as did the dry region equation. This is to be expected, since one expects a better correlation when fewer variables are involved. The correlation is, however, satisfactory with heat transfer and condensation rates correlated within 10% of three region model predictions in the range of inlet conditions shown in Table 5.9. The leaving air temperature predictions are very close between the two models. The leaving air was predicted by both models to be saturated for the deep six row coil modelled.

The agreement between the parametric and the three region model predictions was poorest for the frosted coil. The verification of the parametric model for the frosted coil is illustrated with five time simulations in Table 5.10. The reference state was selected as:

$$T_a = 32 \text{ °F}$$
$$T_r = 10 \text{ °F}$$
$$\varphi = 0.80$$
$$V_{fc} = 300 \text{ fpm}$$
$$x_{in} = 0.20$$

and the parametric equations (4.33), and (4.34) were determined as:

$$\Delta CC = -0.40755 \cdot \Delta T_{a,r} + 0.03345 \cdot \Delta T_{a,r}^2 - 3.2733 \cdot \Delta\varphi + 0.01074 \cdot \Delta V_{fc} - 2.57 \cdot \Delta x_{in} \qquad (5.5)$$

$$\Delta \varepsilon_i = 0.022 \cdot \Delta T_{a,r} + 0.32 \cdot \Delta\varphi + 8.0 \cdot 10^{-5} \cdot \Delta V_{fc} + 0.03 \cdot \Delta x_{in} \qquad (5.6)$$

$$\text{If } \varepsilon_i > 1.0 \text{ , set } \varepsilon_i = 1.0 \qquad (5.6.1)$$

The thermal resistance of the frost layer is not accounted for in the parametric model. The performance deterioration due to frost is mainly due to the reduced air mass flow which is accounted for by the parametric model in the manner described in section 4.4. It is clear from the simulations that the processes predicted by the parametric and the three region models follow the same trend. The discrepancies between the two models are small at the outset but grow with time, which is to be expected. However, the deviations in the predicted air mass flow rate grow unexpectedly large in some cases. This can not be explained by the fact that the frost thermal resistance is not included in the parametric model. It was pointed out previously that any discrepancy in the predicted dehumidification rate can have significant effects on the results due to the sensitivity of the frost property correlation on the latent to sensible heat transfer ratio. In the simulations of Table 5.10 the parametric model overestimates the dehumidification rates for the first hours. As a result calculated frost density is lower for the parametric model than the three region model and therefore the flow is more obstructed in the parametric model simulation which is confirmed by the consistent underestimate of the air mass flow rate in the parametric model relative to the three region model, at the end of the simulations.

It is clear that the frosted coil simulation is very sensitive to initial conditions and calculated conditions such as heat transfer rates, dehumidification rates, and surface temperatures. All frosted coil models, however, show the expected trends in all calculated variables, as is evident from Figures 5.3, 5.4, and 5.5, and Table 5.10. It should be kept in mind that the parametric model is to be used in a heat pump model where the effects of various variables (parameters in the equations (4.33) (4.34), and (4.35)) on the heat pump performance are to be studied.

INPUT	OPERATING POINTS							
	1	2	3	4	5			
T_a	55.4	48.4	41.2	53.4	64.8			
φ	0.69	0.68	0.74	0.87	0.45			
x_{in}	0.28	0.19	0.22	0.26	0.26			
T_{ev}	28.3	27.1	21.7	28.8	35.2			
m_s	74.1	68.6	60.2	68.4	71.1			
F.E.M.								
q	21762	17780	15935	21023	21102			
$\dot{m}_v$	4.01	1.49	1.93	7.70	0.0			
T_{a2}	42.9	37.0	31.6	44.4	49.4			
φ	0.98	1.00	1.00	1.00	0.78			
ΔT_{sup}	5.3	---	---	12.8	5.96			
x_{out}	---	0.848	0.885	---	---			
3-REG.								
q	21798	17523	15841	20987	21221			
$\dot{m}_v$	4.07	1.40	1.89	7.72	0.0			
T_{a2}	43.0	37.1	31.6	44.4	49.3			
φ	0.98	1.00	1.00	1.00	0.79			
ΔT_{sup}	12.3	---	---	16.1	12.3			
x_{out}	---	0.843	0.877	---	---			
ERROR%								
q	+ 0.2	- 1.4	- 0.6	- 0.2	+ 0.6			
$\dot{m}_v$	+ 0.1	- 6.3	- 2.1	+ 0.2	0.0			
ΔT_{sup}	+132.	---	---	+25.6	+106.			
x_{out}	---	- 0.6	- 0.9	---	---			
ERROR								
ΔT_{a2}	0.1	0.1	0.0	0.0	- 0.1			
$\Delta\varphi$	0.0	0.0	0.0	0.0	0.01			

TABLE 5.1

Comparison of Finite Element and Three Region Model Predictions on Dry and Wet Six Row Coil.

INPUT	TIME							
	2.0	4.0	6.0		0.5	1.0	1.5	2.0
T_a	32.4	32.4	32.4		35.0	35.0	35.0	35.0
φ	0.81	0.81	0.81		0.75	0.75	0.75	0.75
x_{in}	0.27	0.27	0.27		0.10	0.10	0.10	0.10
T_{ev}	11.7	11.7	11.7		14.0	14.0	14.0	14.0
m_r	46.6	46.6	46.6		31.6	31.6	31.6	31.6
F.E.M.								
q	14436	14321	14274		11720	11719	11718	11718
m_v	2.75	2.79	2.80		1.32	1.33	1.34	1.36
m_{fst}	5.48	11.0	16.6		0.66	1.33	2.00	2.68
δ_{fst}	.0134	.0213	.0253		.0011	.0021	.0030	.0038
T_{a2}	24.6	24.7	24.7		27.6	27.6	27.6	27.6
φ	0.99	0.99	0.98		0.96	0.96	0.96	0.96
ΔT_{sup}	9.8	10.0	5.7		5.8	5.8	5.8	5.8
3-REG.								
q	14100	14091	14076		11775	11771	11771	11771
m_v	2.65	2.65	2.64		1.15	1.15	1.15	1.17
m_{fst}	5.29	10.6	15.9		0.58	1.15	1.73	2.31
δ_{fst}	.0114	.0183	.0231		.0010	.0019	.0028	.0036
T_{a2}	24.7	24.7	24.7		27.4	27.4	27.4	27.4
φ	0.99	0.99	0.99		0.98	0.98	0.98	0.98
ΔT_{sup}	8.1	7.8	7.5		10.4	10.4	10.4	10.4
ERROR%								
q	− 2.3	− 1.6	− 1.4		+ 0.5	+ 0.4	+ 0.5	+ 0.5
m_v	− 3.8	− 5.0	− 5.8		−12.7	−13.4	−14.2	−14.1
m_{fst}	− 3.5	− 4.1	− 4.6		−12.6	−13.0	−13.4	−13.6
δ_{fst}	−14.9	−14.1	− 8.7		− 8.8	− 8.6	− 6.0	− 5.7
ΔT_{sup}	−17.4	−22.0	+31.6		+80.0	+79.5	+79.1	+79.1
ERROR								
ΔT_{a2}	0.1	0.0	0.0		− 0.2	− 0.2	− 0.2	− 0.2
$\Delta\varphi$	0.0	0.0	0.01		0.02	0.02	0.02	0.02

TABLE 5.2

Comparison of Finite Element and Three Region Model Predictions on Frosted Six Row Coil.

INPUT	OPERATING POINTS								
	1	2	3	4	5	6	7	8	9
T_a	65.8	55.4	53.4	48.4	41.2	53.4	64.8	52.7	51.0
φ	0.45	0.69	0.52	0.68	0.74	0.87	0.45	0.84	0.85
x_{in}	0.25	0.28	0.27	0.19	0.22	0.26	0.26	0.28	0.27
T_{ev}	34.6	28.3	27.3	27.1	21.7	28.8	35.2	27.8	27.4
x_{out}	----	----	----	0.843	0.877	----	----	----	----
ΔT_{sup}	20.9	12.3	9.81	----	----	16.1	12.3	11.9	12.8
3-REG.									
q	19996	21798	17964	17523	15841	20987	21221	22939	20070
m_v	0.0	4.07	0.0	1.40	1.89	7.72	0.0	7.82	6.61
T_{a2}	51.2	43.0	40.6	37.1	31.6	44.4	49.3	42.4	41.8
φ	0.76	0.98	0.84	1.00	1.00	1.00	0.79	1.00	1.00
m_r	64.2	74.1	60.4	68.6	60.2	68.4	71.1	77.9	66.6
EXPER.									
q	20383	19848	16140	17007	15433	19283	19060	19703	18414
m_v	0.0	4.01	0.13	2.32	2.94	7.32	0.44	7.13	6.16
T_{a2}	52.6	44.6	43.6	37.9	32.4	44.5	53.9	43.7	41.8
φ	0.73	0.93	0.74	0.93	0.92	0.99	0.66	0.96	1.00
m_r	65.4	67.4	54.3	66.4	58.6	62.9	63.9	66.9	61.2
ERROR%									
q	− 1.8	+ 9.8	+11.3	+ 3.0	+ 2.6	+ 8.8	+11.3	+16.4	+ 9.0
m_v	0.0	+ 1.5	----	−39.7	−35.7	+ 5.5	----	+ 9.6	+ 7.3
m_r	− 1.8	+ 9.9	+11.2	+ 3.3	+ 2.7	+ 8.7	+11.3	+14.1	+ 8.8
ERROR									
ΔT_{a2}	− 1.4	− 1.6	− 3.0	− 0.8	− 0.8	− 0.1	− 4.6	− 1.3	0.0
$\Delta\varphi$	0.03	0.05	0.10	0.07	0.08	0.01	0.13	0.04	0.0

TABLE 5.3

Comparison of Three Region Model Predictions with Experimental Results on Dry and Wet Six Row Coil.

INPUT	OPERATING POINTS								
	10	11	12	13					
T_a	46.6	48.2	48.6	31.6					
φ	0.74	0.63	0.62	0.36					
x_{in}	0.28	0.21	0.19	0.16					
T_{ev}	19.8	25.5	27.0	14.7					
x_{out}	—	0.858	0.840	0.77					
ΔT_{sup}	12.3	—	—	—					
3-REG.									
q	21972	18274	16893	11760					
$\dot{m}_v$	4.65	0.54	0.0	0.0					
T_{a2}	34.7	35.7	36.7	23.6					
φ	1.00	1.00	0.98	0.52					
$\dot{m}_r$	73.2	71.7	66.5	48.1					
EXPER.									
q	18935	17779	16978	13204					
m_v	4.04	1.97	1.36	0.56					
T_{a2}	36.9	38.0	38.1	24.4					
φ	0.94	0.87	0.89	0.47					
m_r	63.0	69.7	66.4	53.7					
ERROR									
q	+16.0	+ 2.8	− 0.5	−10.9					
m_v	+15.0	−72.6	—	—					
m_r	+16.2	+ 2.9	+ 0.1	−10.4					
ERROR									
ΔT_{a2}	− 2.2	− 2.3	− 1.4	− 0.8					
$\Delta\varphi$	0.06	0.13	0.09	0.05					

TABLE 5.3 (continued)

INPUT	OPERATING POINTS								
	1	2	3	4	5	6	7	8	9
T_a	38.5	39.1	39.6	37.9	34.4	43.7	37.9	35.2	46.2
φ	0.71	0.56	0.55	0.64	0.50	0.40	0.71	0.52	0.36
x_{in}	0.08	0.08	0.08	0.07	0.06	0.06	0.15	0.14	0.12
T_{ev}	22.7	23.4	23.7	23.4	20.1	26.8	21.7	21.9	29.6
x_{out}	0.820	0.791	0.843	0.797	0.798	0.793	0.960	0.910	0.910
ΔT_{sup}	——	——	——	——	——	——	——	——	——
3-REG.									
q	6423	6444	6391	5617	5485	7113	6008	4603	6661
m_v	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0
T_{a2}	36.6	37.2	37.7	36.2	32.8	41.6	36.1	33.8	44.2
φ	0.77	0.60	0.59	0.68	0.53	0.43	0.76	0.55	0.39
m_r	96.5	100.9	93.3	86.4	82.6	108.9	82.4	66.5	95.0
EXPER.									
q	6720	6781	6657	6533	6224	6851	6677	5593	6578
m_v	2.28	0.87	0.82	0.84	0.71	0.69	0.0	0.0	0.25
T_{a2}	38.0	36.0	36.5	35.2	31.7	39.7	37.5	33.2	42.6
φ	0.71	0.61	0.59	0.69	0.53	0.45	0.73	0.57	0.41
m_r	100.5	105.3	97.0	100.3	93.4	104.3	91.6	81.0	93.9
ERROR%	%	%	%	%	%	%	%	%	%
q	− 3.9	− 4.9	− 4.0	−14.0	−11.9	+ 3.8	−10.0	−17.7	+ 1.2
m_v	——	——	——	——	——	——	0.0	0.0	——
m_r	− 3.9	− 4.2	− 3.8	−13.8	−11.6	+ 4.4	−10.0	−17.9	+ 1.3
ERROR									
ΔT_{a2}	− 1.4	1.2	1.2	1.0	1.1	1.9	− 1.4	0.6	1.6
$\Delta\varphi$	0.06	−0.01	0.0	−0.01	0.0	−0.02	0.03	−0.02	−0.02

TABLE 5.4

Comparison of Three Region Model Predictions with Experimental Results on One Row Coil

INPUT	TIME (hrs)								
	1.0	2.0	3.0	4.0	5.0	6.0	7.0	8.0	9.0
T_a	34.0	34.3	34.9	36.4	36.0	36.8	37.4	36.1	36.3
φ	0.83	0.80	0.75	0.71	0.71	0.77	0.79	0.83	0.76
x_{in}	0.19	0.19	0.18	0.19	0.19	0.21	0.19	0.18	0.18
P_{ev}	33.6	33.5	33.7	34.2	33.7	34.2	34.6	34.2	33.5
x_{out}	0.839	0.841	0.833	0.841	0.861	0.881	0.851	0.837	0.836
ΔT_{sup}	---	---	---	---	---	---	---	---	---
3-REG.									
q	14370	14510	14176	14395	14738	16179	16939	16856	17660
m_a	6031	6031	6028	6022	6004	6009	5996	5987	6003
m_v	2.99	2.57	1.65	1.01	1.16	2.65	3.28	3.90	3.02
m_{fst}	2.99	5.77	7.42	8.43	9.60	12.2	15.5	19.4	22.5
T_{a2}	26.3	26.2	26.4	27.3	26.7	27.6	28.1	27.4	26.4
φ	1.00	1.00	1.00	1.00	1.00	1.00	1.00	1.00	1.00
m_r	55.0	55.0	55.0	55.0	55.0	60.5	64.0	64.0	67.3
EXPER.									
q	13953	13970	13975	14364	14417	14696	14756	14487	14173
m_a	5626	5529	5541	5732	5725	5540	5492	5342	5291
m_v	3.79	3.36	2.63	2.23	2.37	3.25	3.44	4.00	3.01
m_{fst}	4.00	7.00	9.10	11.4	14.6	18.0	22.0	25.4	28.3
T_{a2}	26.7	26.5	26.6	27.8	27.4	28.5	29.0	28.2	27.8
φ	0.92	0.92	0.92	0.90	0.90	0.92	0.93	0.93	0.92
m_r	53.4	53.4	53.7	54.8	54.0	54.5	55.7	55.2	54.0
ERROR%									
q	+ 3.0	+ 3.9	+ 1.4	+ 0.2	+ 2.1	+10.1	+14.8	+16.3	+24.6
m_a	+ 7.2	+ 9.1	+ 8.8	+ 5.1	+ 4.9	+ 8.5	+ 9.2	+12.1	+13.4
m_v	−21.1	−23.5	−37.1	−54.6	−50.9	−18.5	− 4.8	− 2.4	+ 0.4
m_{fst}	−25.2	−17.5	−18.4	−26.0	−34.3	−32.0	−29.4	−23.5	−20.6
m_r	+ 3.0	+ 3.0	+ 2.4	+ 0.4	+ 1.8	+11.0	+14.9	+15.9	+24.6
ERROR									
ΔT_{a2}	−0.37	−0.27	−0.18	−0.50	−0.70	−0.87	−0.87	−0.85	−1.43
$\Delta\varphi$	0.08	0.08	0.08	0.10	0.10	0.08	0.07	0.07	0.08

TABLE 5.5

Comparison of Three Region Model Predictions with Experimental Results on Frosted Six Row Coil. Simulation A.

INPUT	TIME (hrs)								
	10.0	11.0	12.0	13.0	14.0	15.0			
T_a	34.9	34.3	33.1	32.1	31.8	32.1			
φ	0.76	0.76	0.78	0.79	0.83	0.84			
x_{in}	0.18	0.18	0.18	0.19	0.18	0.19			
P_{ev}	32.7	32.2	31.5	30.7	30.2	29.9			
x_{out}	0.834	0.822	0.824	0.830	0.824	0.824			
ΔT_{sup}	—	—	—	—	—	—			
3-REG.									
q	17544	17435	16682	16389	15995	14564			
m_a	6003	5642	5070	4588	3878	3044			
m_v	2.99	3.12	3.38	3.54	4.00	3.81			
m_{fst}	25.4	28.6	31.9	35.5	39.4	43.2			
T_{a2}	25.0	24.0	22.4	20.7	19.3	17.8			
φ	1.00	1.00	1.00	1.00	1.00	1.00			
m_r	67.3	67.3	64.4	63.0	61.4	56.2			
EXPER.									
q	13713	13368	12970	12522	12157	11889			
m_a	5298	5172	5052	4800	4354	3867			
m_v	2.87	2.83	3.00	2.97	3.27	3.43			
m_{fst}	31.2	34.2	37.2	40.5	43.8	47.2			
T_{a2}	26.6	26.0	25.1	24.1	23.6	23.3			
φ	0.91	0.91	0.91	0.91	0.92	0.92			
m_r	52.4	51.4	50.0	48.1	46.9	46.1			
ERROR%									
q	+27.9	+30.4	+28.6	+30.9	+31.6	+22.5			
m_a	+13.3	+ 9.1	+ 0.4	− 4.4	−10.9	−21.3			
m_v	+ 4.1	+10.2	+12.6	+19.1	+22.2	+11.0			
m_{fst}	−18.5	−16.5	−14.2	−12.4	− 9.9	− 8.4			
m_r	+28.4	+30.9	+28.8	+31.0	+30.9	+21.9			
ERROR									
ΔT_{a2}	−1.58	−2.04	−2.67	−3.36	−4.32	−5.51			
$\Delta\varphi$	0.09	0.09	0.09	0.09	0.08	0.08			

TABLE 5.5 (continued)

INPUT	TIME (hrs)							
	1.0	2.0	3.0	4.0	5.0	6.0	7.0	8.0
T_e	27.2	27.4	27.6	28.4	28.4	28.2	27.9	27.8
φ	0.91	0.90	0.91	0.91	0.91	0.89	0.88	0.85
x_{in}	0.19	0.19	0.18	0.21	0.19	0.21	0.21	0.19
P_{ev}	31.1	31.0	31.5	31.2	31.5	30.8	30.2	29.6
x_{out}	0.829	0.844	0.812	0.867	0.833	0.878	0.870	0.831
ΔT_{sup}	—	—	—	—	—	—	—	—
3-REG.								
q	10646	11139	10423	12243	11863	12760	13845	13784
m_a	6122	6122	6119	6116	6105	6105	6108	5149
m_v	2.50	2.67	2.47	3.10	2.99	2.99	3.17	2.95
m_{fst}	2.51	5.09	7.56	10.7	13.6	16.6	19.8	22.7
T_{at}	21.8	21.8	22.3	22.2	22.4	21.7	20.7	19.2
φ	1.00	1.00	1.00	1.00	1.00	1.00	1.00	1.00
m_r	41.2	42.4	40.6	45.9	45.9	47.6	52.3	52.3
EXPER.								
q	12508	12647	12575	12904	12806	12686	12368	11867
m_a	6512	6403	6261	5987	5691	5057	4579	4165
m_v	3.50	3.51	3.48	3.67	3.71	3.49	3.34	2.93
m_{fst}	2.10	5.60	9.20	13.0	16.5	19.9	23.0	25.6
T_{at}	21.6	21.6	21.7	22.2	22.0	20.9	19.9	19.1
φ	0.95	0.96	0.96	0.96	0.96	0.95	0.95	0.94
m_r	48.2	48.2	49.0	48.4	49.1	47.3	46.2	45.4
ERROR%								
q	−14.9	−11.9	−17.1	−5.12	−7.36	+0.58	+11.9	+16.1
m_a	− 6.0	− 4.4	− 2.3	+ 2.1	+ 7.3	+20.0	+33.3	+23.6
m_v	−28.0	−24.0	−29.0	−15.5	−19.4	−14.3	− 5.1	+ 0.6
m_{fst}	+19.0	− 9.1	−17.8	−17.7	−17.6	−16.6	−13.9	−11.3
m_r	−14.5	−12.0	−17.1	− 5.2	− 6.5	+ 0.6	+13.2	+15.2
ERROR								
ΔT_{a2}	0.20	0.20	0.60	0.00	0.40	0.80	0.80	0.10
$\Delta \varphi$	0.05	0.04	0.04	0.04	0.04	0.05	0.05	0.06

TABLE 5.6

Comparison of Three Region Model Predictions with Experimental Results on Frosted Six Row Coil. Simulation B.

INPUT	TIME (hrs)							
	9.0	10.0						
T_a	27.4	27.2						
φ	0.82	0.80						
x_{in}	0.17	0.18						
P_{ev}	29.0	28.1						
x_{out}	0.805	0.821						
ΔT_{sup}	---	---						
3-REG.								
q	13397	13434						
m_a	4476	3816						
m_v	2.64	2.60						
m_{fst}	25.4	28.1						
T_{a2}	17.6	17.9						
φ	1.00	1.00						
m_s	52.3	51.8						
EXPER.								
q	11425	11036						
m_a	3947	3780						
m_v	2.55	2.35						
m_{fst}	28.2	30.3						
T_{a2}	18.3	17.8						
φ	0.93	0.93						
m_s	44.6	42.3						
ERROR%								
q	+17.3	+21.7						
m_a	+13.4	+ 0.9						
m_v	+ 3.5	+10.6						
m_{fst}	− 9.9	− 7.3						
m_s	+17.3	+22.3						
ERROR								
ΔT_{a2}	0.70	0.10						
$\Delta\varphi$	0.07	0.07						

TABLE 5.6 (continued)

INPUT	TIME (hrs)								
	1.0	2.0	3.0	4.0	5.0	6.0	7.0	8.0	9.0
T_a	35.8	35.3	34.9	34.0	32.7	32.1	31.3	30.6	30.4
φ	0.75	0.75	0.75	0.77	0.80	0.81	0.83	0.84	0.85
x_{in}	0.27	0.27	0.27	0.27	0.27	0.27	0.27	0.27	0.27
P_{ev}	31.7	31.2	31.4	31.5	30.9	30.7	30.2	30.3	29.4
x_{out}	—	—	—	—	—	—	—	—	—
ΔT_{sup}	11.3	12.4	11.0	10.7	9.01	7.81	8.20	5.39	5.94
3-REG.									
q	14227	14282	13525	12316	12702	12813	13011	12284	14409
m_a	6010	6010	6017	6022	6034	6050	6058	6068	6077
m_v	1.87	1.94	1.68	1.57	2.10	2.24	2.52	2.36	3.12
m_{fst}	1.87	3.78	5.45	7.03	9.13	11.3	13.9	16.2	19.3
T_{a2}	27.4	26.9	26.8	26.7	25.5	24.9	24.3	23.9	22.9
φ	0.98	0.97	0.98	0.98	0.99	0.99	0.99	1.00	1.00
m_r	46.9	46.9	44.6	40.7	42.0	42.5	43.2	40.8	48.0
EXPER.									
q	13185	12670	12604	12469	12105	11667	11656	11328	11010
m_a	6391	6315	6282	6074	6025	5810	5931	5952	5713
m_v	2.13	1.90	1.88	2.18	2.48	2.51	2.65	2.60	2.67
m_{fst}	2.10	3.90	6.00	8.40	10.9	13.5	16.0	18.7	21.2
T_{a2}	28.7	28.3	27.9	27.1	26.2	25.7	25.1	24.7	24.5
φ	0.92	0.92	0.92	0.93	0.94	0.94	0.95	0.94	0.95
m_r	43.5	41.6	41.6	41.2	40.0	39.6	38.7	38.9	37.6
ERROR%									
q	+ 7.9	+12.7	+ 7.3	− 1.2	+ 4.9	+ 9.8	+11.6	+ 8.4	+30.9
m_a	− 6.0	− 4.8	− 4.3	− 1.0	− 0.2	+ 4.1	+ 2.1	+ 1.9	+ 6.4
m_v	−12.2	+ 2.1	−10.6	−27.9	−15.3	−10.8	− 5.0	− 9.2	+16.8
m_{fst}	−10.9	− 3.1	− 9.2	−16.3	−16.2	−16.0	−13.4	−13.2	− 8.8
m_r	+ 7.8	+12.7	+ 7.2	− 1.2	+ 5.0	+ 7.3	+11.6	+ 4.9	+27.7
ERROR									
ΔT_{a2}	−1.30	−1.40	−1.10	−0.40	−0.70	−0.76	−0.84	−0.82	−1.63
$\Delta\varphi$	0.06	0.05	0.06	0.05	0.05	0.05	0.04	0.06	0.05

TABLE 5.7

Comparison of Three Region Model Predictions with Experimental Results on Frosted Six Row Coil. Simulation C.

INPUT	TIME (hrs)							
	10.0	11.0	12.0					
T_a	30.5	28.8	28.5					
φ	0.85	0.87	0.87					
x_{in}	0.27	0.26	0.24					
P_{ev}	28.7	28.2	28.6					
x_{out}	——	——	——					
ΔT_{sup}	6.49	6.77	4.38					
3-REG.	-----	-----	-----	-----	-----	-----	-----	-----
q	16196	14581	12562					
m_a	6080	5899	5377					
m_v	3.67	3.31	2.79					
m_{fst}	23.0	26.3	29.1					
T_{a2}	22.2	21.1	21.0					
φ	1.00	1.00	1.00					
m_r	53.6	47.5	40.2					
EXPER.	-----	-----	-----	-----	-----	-----	-----	-----
q	10475	10171	10179					
m_a	5423	5592	4934					
m_v	2.49	2.51	2.57					
m_{fst}	23.7	26.2	28.8					
T_{a2}	24.6	23.3	22.2					
φ	0.95	0.95	0.95					
m_r	35.8	34.8	36.4					
ERROR%	-----	-----	-----	-----	-----	-----	-----	-----
q	+54.6	+43.3	+23.4					
m_a	+12.1	+ 5.5	+ 9.0					
m_v	+47.4	+31.9	+ 8.7					
m_{fst}	− 2.9	+ 0.5	+ 1.1					
m_r	+49.7	+36.5	+10.4					
ERROR	-----	-----	-----	-----	-----	-----	-----	-----
ΔT_{a2}	−2.45	−2.19	−1.19					
$\Delta\varphi$	0.05	0.05	0.05					

TABLE 5.7 (continued)

INPUT	OPERATING POINTS							
	1	2	3	4	5	6	7	8
T_a	40.0	57.0	60.0	42.0	50.0	55.0	45.0	40.0
φ	0.20	0.45	0.30	0.50	0.40	0.40	0.35	0.45
T_{ev}	20.0	28.0	33.0	17.0	20.0	27.0	22.0	17.0
V_{fc}	350.0	250.0	300.0	350.0	400.0	350.0	300.0	250.0
x_{in}	0.20	0.15	0.30	0.20	0.20	0.10	0.15	0.20
ΔT_{sup}	10.0	10.0	10.0	10.0	10.0	10.0	10.0	10.0
3-REG.								
q	11385	17706	18468	17370	25422	20699	13446	12046
m_v	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0
T_{at}	32.9	41.0	46.0	31.1	35.8	41.7	35.1	29.5
φ	0.27	0.82	0.50	0.79	0.69	0.66	0.51	0.69
PARAM.								
q	11815	17140	18055	17764	25513	20391	13647	12302
m_v	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0
T_{at}	32.6	41.5	46.3	30.9	35.8	41.9	35.0	29.2
φ	0.27	0.80	0.50	0.78	0.69	0.65	0.52	0.70
ERROR%								
q	+ 3.8	− 3.2	− 2.2	+ 2.3	+ 0.4	− 1.5	+ 1.5	+ 2.1
m_v	—	—	—	—	—	—	—	—
ERROR								
ΔT_{at}	− 0.3	0.5	0.3	− 0.2	0.0	0.2	− 0.1	− 0.3
$\Delta\varphi$	0.0	−0.02	0.0	−0.01	0.0	−0.01	0.01	0.01

TABLE 5.8

Comparison of Three Region and Parametric Model Predictions on Dry Six Row Coil.

INPUT	OPERATING POINTS							
	1	2	3	4	5	6	7	8
T_a	60.0	60.0	60.0	45.0	45.0	45.0	55.0	57.0
φ	0.60	0.90	0.80	0.75	0.80	0.90	0.75	0.80
T_{ev}	17.0	26.0	30.0	17.0	22.0	24.0	25.0	30.0
$\dot{V}_{fc}$	200.0	350.0	250.0	250.0	300.0	300.0	350.0	300.0
x_{in}	0.10	0.20	0.15	0.20	0.15	0.20	0.15	0.25
ΔT_{sup}	10.0	10.0	10.0	10.0	10.0	10.0	10.0	10.0
3-REG.								
q	35257	47174	28609	21600	16695	16106	30808	26067
m_v	9.95	21.8	11.0	5.29	3.81	5.39	8.93	8.96
T_{al}	32.2	45.2	45.0	31.0	35.7	37.5	41.4	44.7
φ	1.00	1.00	1.00	1.00	0.99	1.00	1.00	1.00
PARAM.								
q	37107	45066	26139	21543	17626	17850	29103	24526
m_v	10.8	21.5	10.0	5.25	4.14	6.17	8.12	8.28
T_{al}	31.0	45.8	46.1	31.0	35.3	36.8	41.9	45.3
φ	1.00	1.00	1.00	1.00	1.00	1.00	1.00	1.00
ERROR%								
q	+ 5.3	− 4.5	− 8.6	− 0.3	+ 5.5	+10.8	− 5.6	− 5.9
m_v	+ 8.3	− 1.6	− 9.2	− 0.8	+ 8.5	+14.5	− 9.0	− 7.6
ERROR								
ΔT_{al}	− 1.2	0.6	1.1	0.0	− 0.4	− 0.7	0.5	0.6
$\Delta\varphi$	0.0	0.0	0.0	0.0	0.01	0.0	0.0	0.0

TABLE 5.9

Comparison of Three Region and Parametric Model Predictions on Wet Six Row Coil.

INPUT	OPERATING POINTS							
	9	10						
T_a	49.0	53.0						
φ	0.85	0.70						
T_{ev}	26.0	30.0						
V_{fc}	400.0	300.0						
x_{in}	0.15	0.20						
ΔT_{sup}	10.0	10.0						
3-REG.								
q	21527	14985						
m_v	6.44	1.40						
T_{a2}	40.9	43.0						
φ	1.00	0.98						
PARAM.								
q	22409	14734						
m_v	6.78	1.44						
T_{a2}	40.6	43.2						
φ	1.00	0.97						
ERROR%								
q	+ 4.1	- 1.7						
m_v	+ 5.2	+ 3.1						
ERROR:								
ΔT_{a2}	- 0.3	0.2						
$\Delta\varphi$	0.0	-0.01						

TABLE 5.9 (continued)

INPUT	TIME (hrs)								
	0.5	1.5		0.5	2.0	4.0	5.0	7.0	10.0
T_a	35.0	35.0		32.0	32.0	32.0	32.0	32.0	32.0
φ	0.90	0.90		0.90	0.90	0.90	0.90	0.90	0.90
T_{ev}	8.00	8.00		12.0	12.0	12.0	12.0	12.0	12.0
V_{fr}	350.0	350.0		350.0	350.0	350.0	350.0	350.0	350.0
x_{in}	0.20	0.20		0.20	0.20	0.20	0.20	0.20	0.20
ΔT_{sup}	10.0	10.0		10.0	10.0	10.0	10.0	10.0	10.0
3-REG.									
q	26740	25040		14538	14208	13708	12714	10758	8713
m_a	6694	6648		6739	6739	6506	5803	4658	3540
m_v	8.08	7.53		3.91	3.82	3.70	3.46	2.95	2.41
m_{fst}	4.04	12.6		1.96	7.73	15.3	18.7	25.3	32.7
T_{a2}	24.0	24.5		25.7	25.8	25.8	25.6	25.3	24.9
φ	1.00	1.00		1.00	1.00	1.00	1.00	1.00	1.00
PARAM.									
q	27297	26779		14202	14202	13641	12767	10701	7706
m_a	6694	6404		6739	6739	6313	5688	4378	2815
m_v	8.48	8.33		3.85	3.85	3.73	3.52	3.01	2.21
m_{fst}	4.24	12.6		1.93	7.70	15.3	18.9	25.3	32.9
T_{a2}	23.7	23.5		25.8	25.8	25.7	25.5	24.9	24.2
φ	1.00	1.00		1.00	1.00	1.00	1.00	1.00	1.00
ERROR%									
q	+ 2.1	+ 6.9		− 2.3	0.0	− 0.4	+ 0.4	− 0.5	−11.6
m_a	0.0	− 3.7		0.0	0.0	− 3.0	− 2.0	− 6.0	−20.5
m_v	+ 4.9	+10.6		− 1.6	+ 0.8	+ 0.8	+ 1.8	+ 1.9	− 8.2
m_{fst}	+ 4.9	+ 8.5		− 1.4	− 0.4	+ 0.4	+ 1.0	+ 1.6	+ 0.8
ERROR									
ΔT_{a2}	− 0.3	− 1.0		0.1	0.0	− 0.1	− 0.1	− 0.4	− 0.7
$\Delta\varphi$	0.0	0.0		0.0	0.0	0.0	0.0	0.0	0.0

TABLE 5.10

Comparison of Three Region and Parametric Model Predictions on Frosted Six Row Coil.

INPUT	TIME (hrs)								
	0.5	2.0	5.0	7.0	9.0		0.5	2.0	5.0
T_a	25.0	25.0	25.0	25.0	25.0		35.0	35.0	35.0
φ	0.95	0.95	0.95	0.95	0.95		0.75	0.75	0.75
T_{ev}	6.00	6.00	6.00	6.00	6.00		14.0	14.0	14.0
V_{fc}	300.0	300.0	300.0	300.0	300.0		300.0	300.0	300.0
x_{in}	0.20	0.20	0.20	0.20	0.20		0.10	0.10	0.10
ΔT_{sup}	10.0	10.0	10.0	10.0	10.0		10.0	10.0	10.0
3-REG.									
q	11396	11137	10933	9681	8121		11775	11771	11778
m_a	5868	5868	5868	5003	4026		5744	5744	5744
m_v	2.97	2.90	2.83	2.50	2.09		1.15	1.17	1.23
m_{fst}	1.48	5.86	14.4	19.7	24.2		0.58	2.31	5.93
T_{a2}	19.5	19.4	19.5	19.2	19.0		27.4	27.4	28.0
φ	1.00	1.00	1.00	1.00	1.00		0.98	0.98	0.97
PARAM.									
q	11820	11820	11514	9046	6938		11626	11626	11626
m_a	5868	5868	5644	4021	2854		5744	5744	5744
m_v	3.17	3.17	3.09	2.44	1.87		1.07	1.07	1.07
m_{fst}	1.59	6.34	15.8	21.2	25.3		0.53	2.13	5.34
T_{a2}	19.0	19.0	19.0	18.4	17.8		27.4	27.4	27.4
φ	1.00	1.00	1.00	1.00	1.00		0.98	0.98	0.98
ERROR%									
q	+ 3.7	+ 6.1	+ 5.3	- 6.6	-14.6		- 1.3	- 1.2	- 1.3
m_a	0.0	0.0	- 3.8	-19.6	-29.0		0.0	0.0	0.0
m_v	+ 6.7	+ 9.4	+ 9.1	- 2.6	-10.4		- 7.4	- 8.5	-13.4
m_{fst}	+ 7.4	+ 8.2	+ 9.6	+ 7.2	+ 4.5		- 7.4	- 7.9	- 9.9
ERROR									
ΔT_{a2}	- 0.5	- 0.4	- 0.5	- 1.3	- 1.2		0.0	0.0	- 0.6
$\Delta\varphi$	0.0	0.0	0.0	0.0	0.0		0.0	0.0	0.0

TABLE 5.10 (continued)

INPUT	TIME (hrs)							
	9.0	15.0		0.5	2.0	5.0	8.0	10.0
T_{ai}	35.0	35.0		30.0	30.0	30.0	30.0	30.0
φ	0.75	0.75		0.85	0.85	0.85	0.85	0.85
T_{ev}	14.0	14.0		8.00	8.00	8.00	8.00	8.00
$\dot{V}_{fc}$	300.0	300.0		250.0	250.0	250.0	250.0	250.0
x_{in}	0.10	0.10		0.15	0.15	0.15	0.15	0.15
ΔT_{sup}	10.0	10.0		10.0	10.0	10.0	10.0	10.0
3-REG.								
q	11767	11868		13434	13210	12964	12642	10352
m_a	5744	5744		4837	4837	4837	4837	3858
m_v	1.32	1.49		3.06	3.01	2.95	2.86	2.36
m_{fst}	11.1	19.6		1.53	6.07	15.0	23.6	28.6
T_{a2}	27.5	27.6		22.3	21.5	21.6	21.7	21.5
φ	0.97	0.95		1.00	1.00	1.00	1.00	1.00
PARAM.								
q	11626	11626		13672	13672	13672	12187	9287
m_a	5744	5744		4837	4837	4837	4082	2816
m_v	1.07	1.07		3.16	3.16	3.16	2.87	2.24
m_{fst}	9.61	16.0		1.58	6.32	15.8	25.0	30.0
T_{a2}	27.4	27.4		21.2	21.2	21.2	20.7	19.9
φ	0.98	0.98		1.00	1.00	1.00	1.00	1.00
ERROR%								
q	− 1.2	− 2.0		+ 1.8	+ 3.5	+ 5.5	− 3.6	−10.3
m_a	0.0	0.0		0.0	0.0	0.0	−15.6	−27.0
m_v	−19.1	−28.5		+ 3.1	+ 5.1	+ 7.2	+ 0.2	− 5.2
m_{fst}	−13.3	−18.4		+ 3.1	+ 4.1	+ 5.5	+ 6.1	+ 4.8
ERROR								
ΔT_{a2}	− 0.1	− 0.2		− 1.1	− 0.3	− 0.4	− 1.0	− 1.6
$\Delta\varphi$	0.01	0.03		0.0	0.0	0.0	0.0	0.0

TABLE 5.10 (continued)

INPUT	OPERATING POINTS							
	1	2						
T_a	53.4	35.0						
φ'	0.87	0.75						
x_{in}	0.26	0.10						
T_{ev}	28.8	14.0						
m_{ref}	68.4	31.6						
w ΔP								
q	21023	11720						
m_v	7.70	1.32						
T_{a2}	44.4	27.6						
φ	1.00	0.96						
ΔT_{sup}	12.8	5.87						
x_{out}	—	—						
w/o ΔP								
q	20951	11703						
m_v	7.67	1.31						
T_{a2}	44.4	27.6						
φ	1.00	0.96						
ΔT_{sup}	13.0	5.76						
x_{out}	—	—						
ERROR%								
q	− 0.3	− 0.1						
m_v	− 0.4	− 0.5						
ΔT_{sup}	+ 1.4	+ 1.9						
x_{out}	—	—						
ERROR								
ΔT_{a2}	0.0	0.0						
$\Delta\varphi$	0.0	0.0						

TABLE 5.11

Finite Element Model Predictions With and Without Refrigerant Side Pressure Drop Accounted For.

INPUT	SIMULATION		
	A	B	C
T_a	34.4	27.7	32.4
ϕ	0.79	0.88	0.81
x_{in}	0.186	0.192	0.270
P_{ev}	31.6	30.4	30.2
x_{out}	0.843	0.839	---
ΔT_{sup}	---	---	8.64

TABLE 5.12
Specified Average Inlet Conditions For Frosted Coil Simulations.

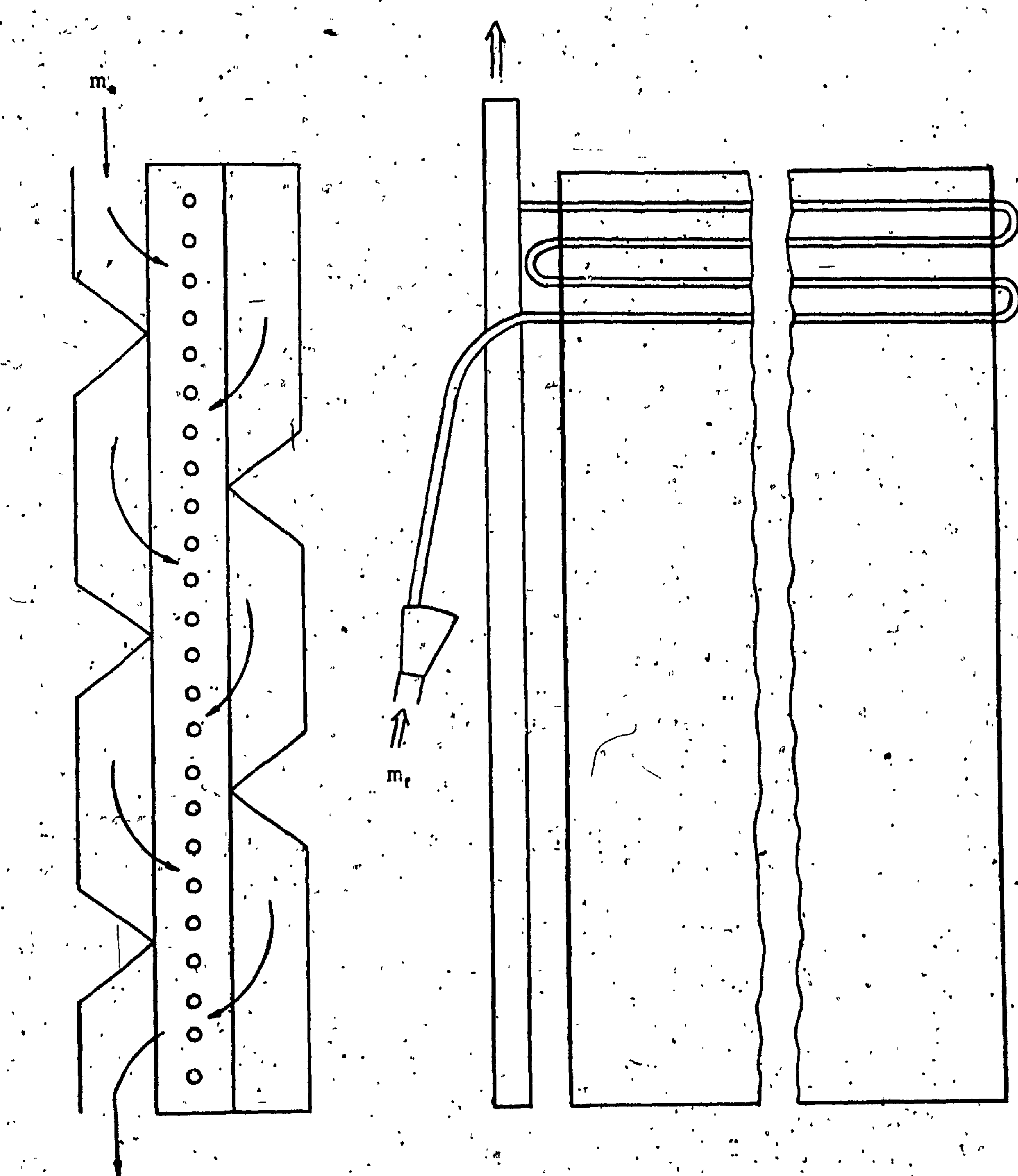


FIGURE 5.1

Six Row Coil

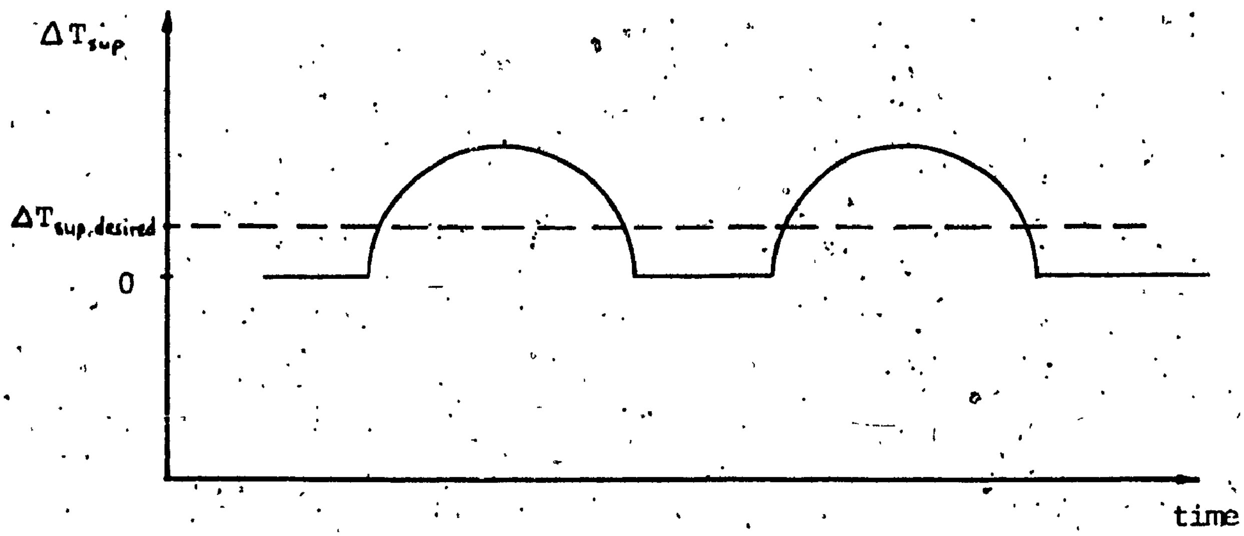
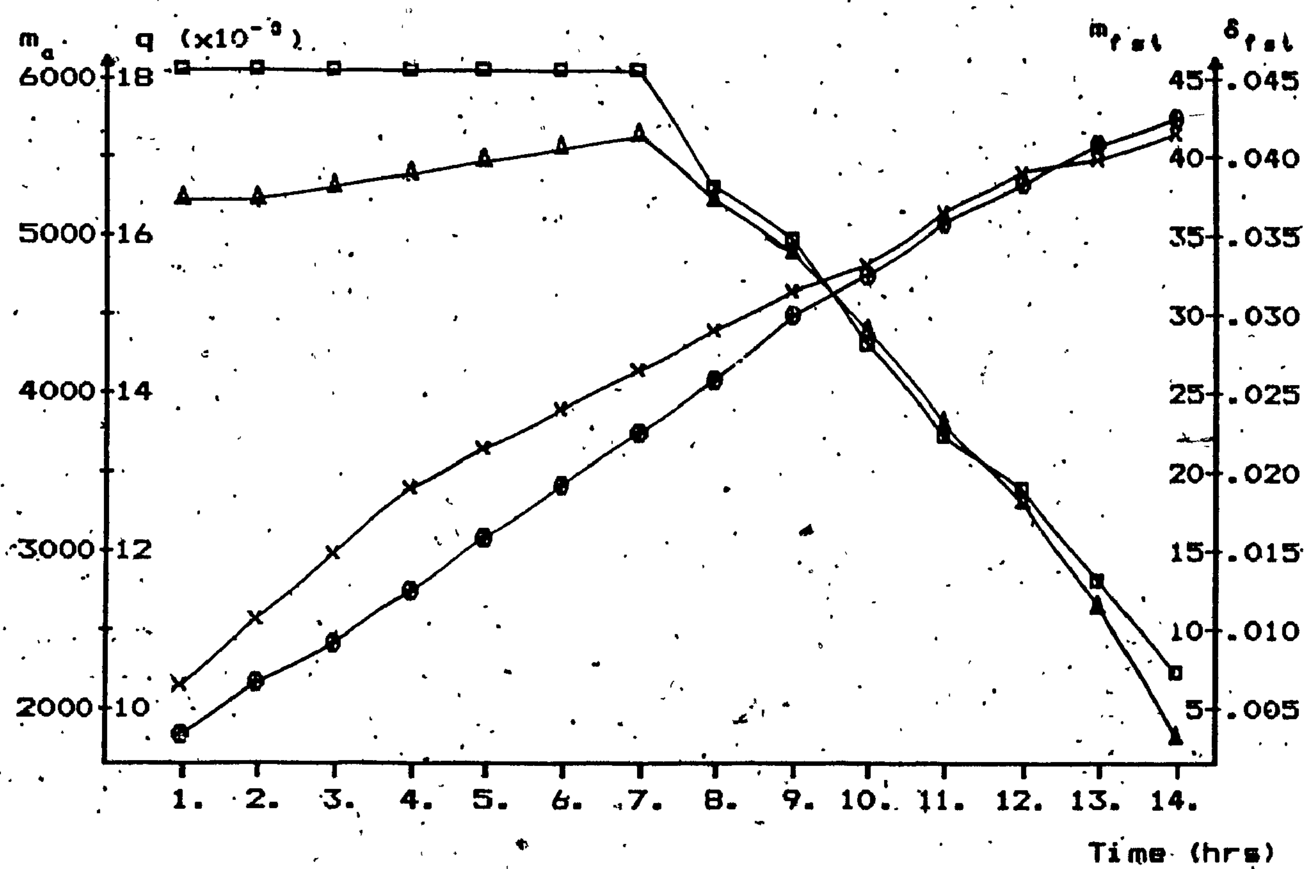


FIGURE 5.2

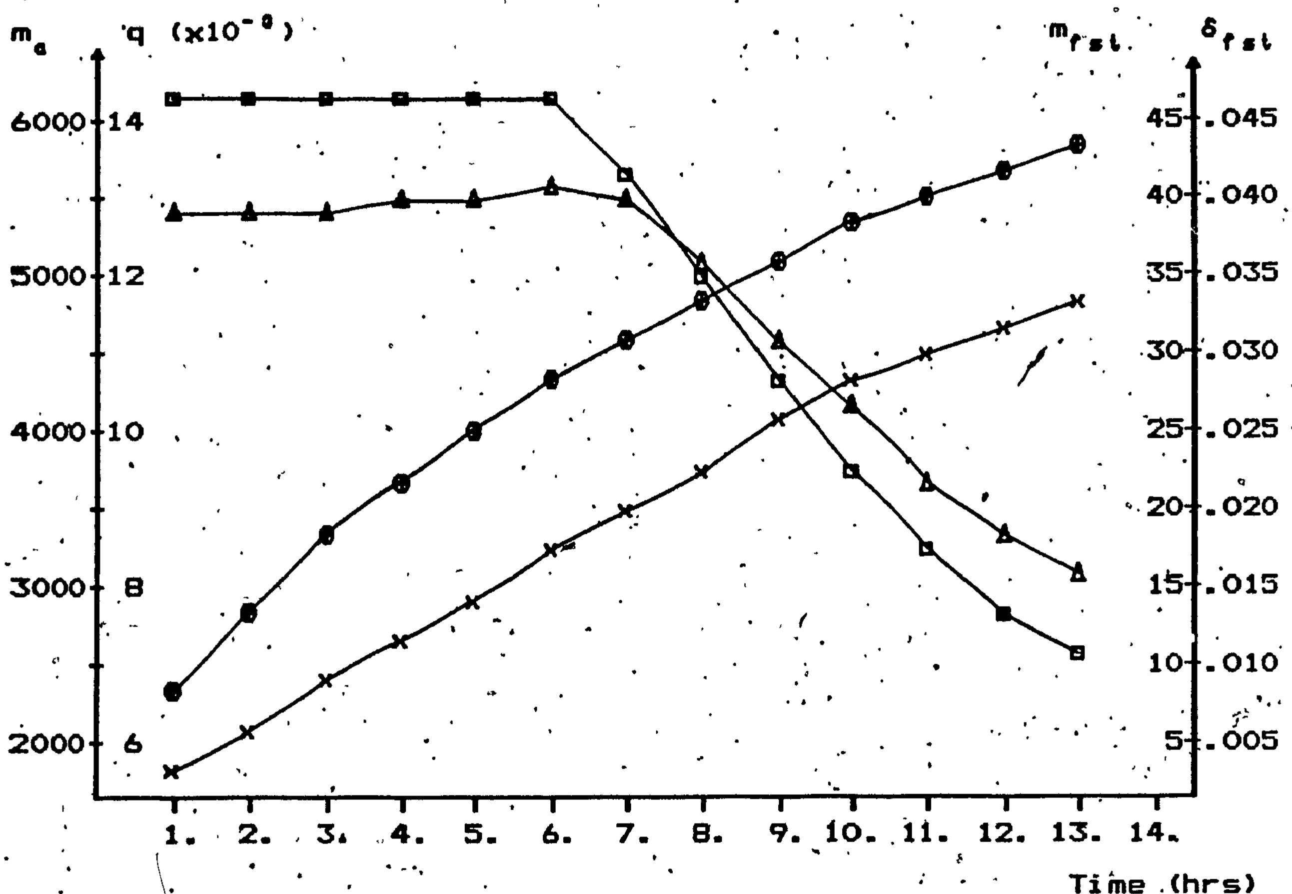
Refrigerant Outlet Degree of Superheat



□ air mass flow; m_a, ($lb_{m,da}$/ hr).
Δ evaporator heat transfer; q, (Btu/hr).
⊕ frost layer thickness; δ_{fst}; (in.)
x accumulated frost; m_{fst}, (lb_m)

FIGURE 5.3

Frosted Six Row Coil Performance as Predicted by Three Region Model Using Average Inlet Conditions in Simulation A.



□ air mass flow; m_a , ($lb_{m,da}$/ hr)

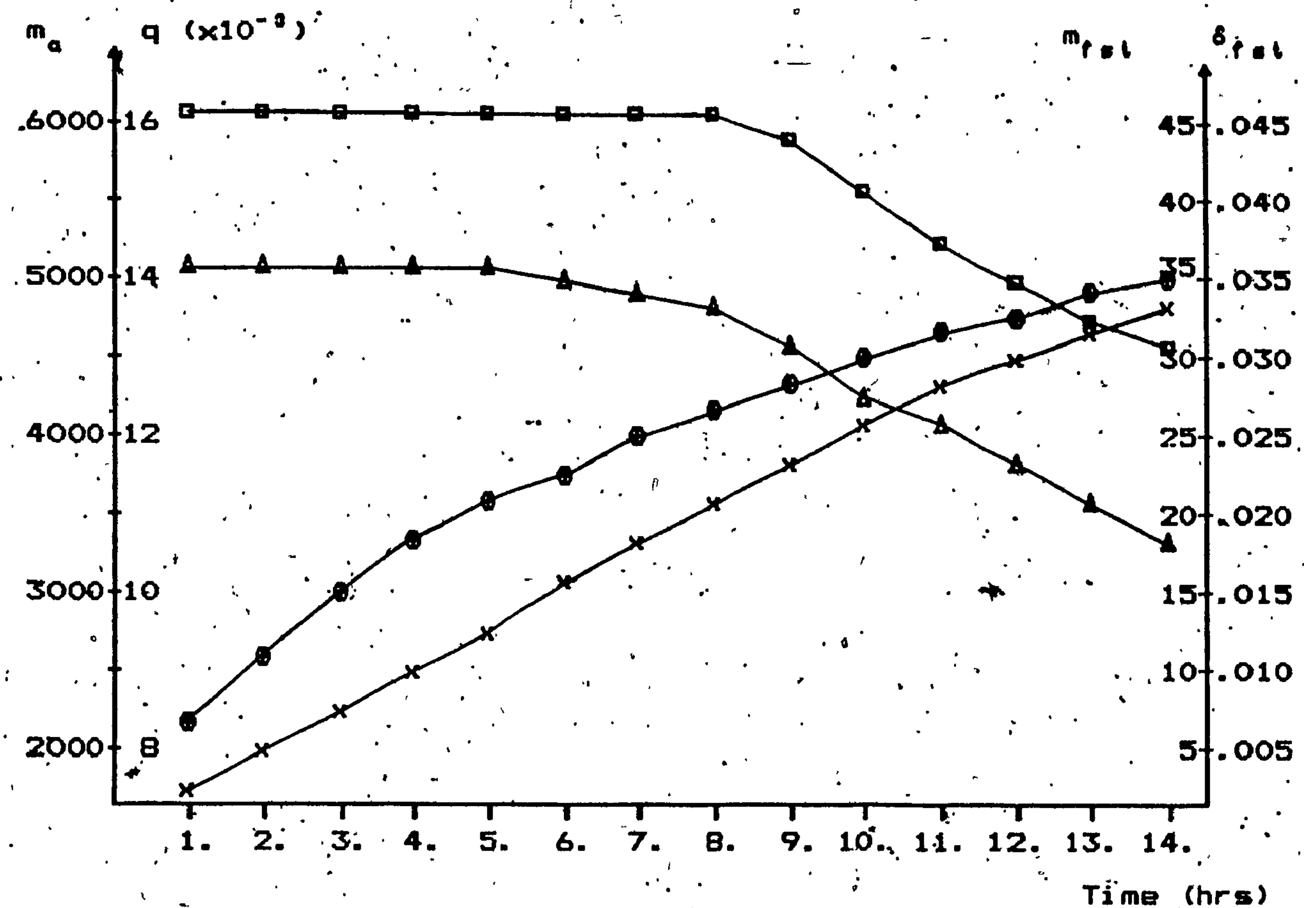
Δ evaporator heat transfer; q , (Btu/hr)

⊕ frost layer thickness; δ_{fst} , (in.)

× accumulated frost; m_{fst} , (lb_m)

FIGURE 5.4

Frosted Six Row Coil Performance as Predicted by Three Region Mode Using Average Inlet Conditions in Simulation B.



□ air mass flow; m_a , ($lb_{m,da}$ / hr)
Δ evaporator heat transfer; q , (Btu/hr)
⊕ frost layer thickness; δ_{fsl} , (in.)
x accumulated frost; m_{fsl} , (lb_m)

FIGURE 5.5
Frosted Six Row Coil Performance as Predicted by Three Region Mode Using Average Inlet Conditions in Simulation C.

CHAPTER 6

CONCLUSIONS

6.1 Immediate Conclusions

Three heat pump evaporator models operating under conditions of dry, wet, and frosted coil surface conditions have been developed:

1. The finite element model. This model allows study of the local behaviour of the heat exchanging fluids and determines refrigerant side pressure drop and its effects on the heat transfer performance of the coil. This effect was found to be negligible. Since the finite element model analyzes the coil in fixed elements the refrigerant quality as a function of position is determined, which allows refrigerant charge to be calculated. Although impractical in actual coil simulations, due to the time consuming calculations required, the finite element model allows evaporator analysis to confirm or repute the validity of various assumptions.

2. The three region model. This model neglects refrigerant side pressure drop, and assumes that no dehumidification takes place in the region of the evaporator carrying superheated refrigerant. These assumptions were found to be very reasonable, by comparing coil performance predictions with the finite element model. The three region model is computationally much faster than the finite element model. The three region model is therefore suitable for use in evaporator research and design work.

3. The parametric model. This model describes the coil performance in terms of the coil characteristic and the coil

effectiveness correlated with two parametric equations. The model was found to be satisfactory. It was found necessary, however, to derive different constants in the parametric equations for the different coil conditions. This is a drawback, but in many cases it will be clear if the dry, wet, or frosted coil equations should be used. This model is short and computationally fast. This feature should be of interest to any heat pump modeller, where computational efficiency is crucial due to the time consuming iterative procedure necessary in heat pump modelling.

6.2 Work direction

The succesful modelling of the frosted coil process will allow heat pump designs used under wet winter conditions to be optimized with respect to the frosting of the outdoor evaporator coil. Frost can not be allowed to build up indefinitely on the evaporator. The frost eventually has to be removed. The defrosting of a coil causes significant heat pump seasonal performance loss. It is therefore desirable to optimize the evaporator performance with respect to frosting and defrosting. The first step in modelling the defrosting of an evaporator is to have a good model for the frosting process of the coil. Good models for the frosted evaporator surface have been provided in this work.

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APPENDIX 1

Sample Output Files

FINITE ELEMENT MODEL OUTPUT

INPUT PARAMETERS

GEOMETRIC DATA. ALL DATA IN UNITS OF FEET.

FACE A.= 4.000 WIDTH= 8.000 HEIGHT= .500 DEPTH= .750

FPI= 8.0 ROWS H.= 4.0 CKTS= 6.0

ROW HEIGHT= .125 ROW DEPTH= .125 DIA.O.= .0422 DIA.I.= .0417

FIN THK.= .00050 FIN SPAC.= .0104 FIN H.= .0414 FINS= 3072.0

ROWS= 6.0 L. CKT= 32.0 KFIN=128.00 BTU/HR*FT*F

ENTERING AIR AND REFRIGERANT STATES

TA1=27.75 F RH= .880 PRA= .718 BTU/LB*F KA= .01 BTU/HR*F*FT

MFA= 6115.2 LB/HR MUA= .0411 LB/FT*HR HFGW=1219.0 BTU/LB

TR=11.77 F HR(1)= 24.02 PR(1)=30.40 PSI MFR= 294.00 LB/HR

SPECIFIED REFR. MASS FLOW= 294.00 LB/HR

THE COIL IS FROSTED

TIME= .50

CALCULATED OVERALL, AND AVERAGE RESULTS

PERCENT OF COIL IN 2 PHASE REF.REGION

SATURATED REGION= 100.0 %

AIR FLOW= 6115.2 LBM/HR   REF.FLOW= 294.0LBM/HR

AVG FROST THK IN 2-PHASE REF.REGION.

AVERAGE THICKNESS= .004533 INCHES   ACC.FROST= 1.441 LBM

HEAT TRANSFER RATES (BTU/HR)

QTOTAL=12992.1   QSENSIBLE= 9479.0
QLATENT= 3513.1

CONDENSATION RATE (LBM/HR)

TOTAL CONDENSATION RATE= 2.882

AVG.QTYS.IN THE 2 PHASE REF.REGION

```
LEAVING DRY BULB = 21.05 F
LEAVING WET BULB = 21.04 F
WALL TEMPERATURE = 21.59 F
REL. HUMIDITY    =  99.9 %
LEAVING HUMIDITY =   .2251E-02 LBWV/LBDA
LATENT ENTHALPY  =  2.39 BTU/LBM
SENSIBLE ENTHALPY=  5.07 BTU/LBM
TOTAL ENTHALPY   =  7.46 BTU/LBM
```

AVG.QTYS. OVER LENGTH OF COIL

```
LEAVING DRY BULB = 21.05 F
LEAVING WET BULB = 21.04 F
WALL TEMPERATURE = 21.59 F
REL. HUMIDITY    =  99.9 %
LEAVING HUMIDITY =   .2251E-02 LBWV/LBDA
LATENT ENTHALPY  =  2.39 BTU/LBM
SENSIBLE ENTHALPY=  5.07 BTU/LBM
TOTAL ENTHALPY   =  7.46 BTU/LBM
```

ENTERING AIR TEMPERATURES

ELEM.	DIST	DR.B.	WT.B.	R.H.	SP.HU.	ENTH	D.PT	S.H.	L.H.	S.V.
1	.00	27.75	26.72	.880	.27E-02	9.6	25.0	6.7	2.89	12.3
2	1.28	27.75	26.72	.880	.27E-02	9.6	25.0	6.7	2.89	12.3
3	2.56	27.75	26.72	.880	.27E-02	9.6	25.0	6.7	2.89	12.3
4	3.84	27.75	26.72	.880	.27E-02	9.6	25.0	6.7	2.89	12.3
5	5.12	27.75	26.72	.880	.27E-02	9.6	25.0	6.7	2.89	12.3
6	6.40	27.75	26.72	.880	.27E-02	9.6	25.0	6.7	2.89	12.3
7	7.68	27.75	26.72	.880	.27E-02	9.6	25.0	6.7	2.89	12.3
8	8.96	27.75	26.72	.880	.27E-02	9.6	25.0	6.7	2.89	12.3
9	10.24	27.75	26.72	.880	.27E-02	9.6	25.0	6.7	2.89	12.3
10	11.52	27.75	26.72	.880	.27E-02	9.6	25.0	6.7	2.89	12.3
11	12.80	27.75	26.72	.880	.27E-02	9.6	25.0	6.7	2.89	12.3
12	14.08	27.75	26.72	.880	.27E-02	9.6	25.0	6.7	2.89	12.3
13	15.36	27.75	26.72	.880	.27E-02	9.6	25.0	6.7	2.89	12.3
14	16.64	27.75	26.72	.880	.27E-02	9.6	25.0	6.7	2.89	12.3
15	17.92	27.75	26.72	.880	.27E-02	9.6	25.0	6.7	2.89	12.3
16	19.20	27.75	26.72	.880	.27E-02	9.6	25.0	6.7	2.89	12.3
17	20.48	27.75	26.72	.880	.27E-02	9.6	25.0	6.7	2.89	12.3
18	21.76	27.75	26.72	.880	.27E-02	9.6	25.0	6.7	2.89	12.3
19	23.04	27.75	26.72	.880	.27E-02	9.6	25.0	6.7	2.89	12.3
20	24.32	27.75	26.72	.880	.27E-02	9.6	25.0	6.7	2.89	12.3
21	25.60	27.75	26.72	.880	.27E-02	9.6	25.0	6.7	2.89	12.3
22	26.88	27.75	26.72	.880	.27E-02	9.6	25.0	6.7	2.89	12.3
23	28.16	27.75	26.72	.880	.27E-02	9.6	25.0	6.7	2.89	12.3
24	29.44	27.75	26.72	.880	.27E-02	9.6	25.0	6.7	2.89	12.3
25	30.72	27.75	26.72	.880	.27E-02	9.6	25.0	6.7	2.89	12.3

L E A V I N G   A I R   T E M P E R A T U R E S

ELEM.	DIST	DR.B.	WT.B.	R.H.	SP.HU.	ENTH	D.PT	S.H.	L.H.	S.V.
1	.00	21.07	21.07	1.000	.23E-02	7.5	21.1	5.1	2.39	12.2
2	1.28	21.07	21.07	1.000	.23E-02	7.5	21.1	5.1	2.39	12.2
3	2.56	21.06	21.06	1.000	.23E-02	7.5	21.1	5.1	2.39	12.2
4	3.84	21.06	21.06	1.000	.23E-02	7.5	21.1	5.1	2.39	12.2
5	5.12	21.05	21.05	1.000	.23E-02	7.5	21.0	5.1	2.39	12.2
6	6.40	21.05	21.05	1.000	.23E-02	7.5	21.0	5.1	2.39	12.2
7	7.68	21.04	21.04	1.000	.23E-02	7.5	21.0	5.1	2.39	12.2
8	8.96	21.04	21.03	1.000	.23E-02	7.5	21.0	5.1	2.39	12.2
9	10.24	21.03	21.03	1.000	.23E-02	7.5	21.0	5.1	2.39	12.2
10	11.52	21.02	21.02	1.000	.23E-02	7.5	21.0	5.1	2.39	12.2
11	12.80	21.02	21.01	1.000	.22E-02	7.5	21.0	5.1	2.39	12.2
12	14.08	21.01	21.01	1.000	.22E-02	7.4	21.0	5.1	2.39	12.2
13	15.36	21.00	21.00	1.000	.22E-02	7.4	21.0	5.1	2.39	12.2
14	16.64	20.99	20.99	1.000	.22E-02	7.4	21.0	5.1	2.38	12.2
15	17.92	20.99	20.98	1.000	.22E-02	7.4	21.0	5.1	2.38	12.2
16	19.20	20.98	20.97	1.000	.22E-02	7.4	21.0	5.1	2.38	12.2
17	20.48	20.97	20.97	1.000	.22E-02	7.4	21.0	5.1	2.38	12.2
18	21.76	20.96	20.96	1.000	.22E-02	7.4	21.0	5.1	2.38	12.2
19	23.04	20.95	20.95	1.000	.22E-02	7.4	20.9	5.0	2.38	12.2
20	24.32	20.94	20.94	1.000	.22E-02	7.4	20.9	5.0	2.38	12.2
21	25.60	20.93	20.93	1.000	.22E-02	7.4	20.9	5.0	2.38	12.2
22	26.88	20.92	20.92	1.000	.22E-02	7.4	20.9	5.0	2.38	12.2
23	28.16	20.96	20.96	1.000	.22E-02	7.4	21.0	5.1	2.38	12.2
24	29.44	21.17	21.17	1.000	.23E-02	7.5	21.2	5.1	2.40	12.2
25	30.72	21.54	21.54	1.000	.23E-02	7.6	21.5	5.2	2.45	12.2

REFRIGERANT TEMPERATURES

DISTANCE	QUALITY	TEMP.	PRES.	ENTH.	REF.VEL.
.00	.192	11.77	30.40	24.02	152.6
1.28	.218	11.76	30.40	25.77	172.4
2.56	.244	11.76	30.39	27.53	192.3
3.84	.270	11.75	30.38	29.29	212.1
5.12	.296	11.74	30.38	31.05	232.0
6.40	.322	11.73	30.37	32.81	251.9
7.68	.349	11.72	30.37	34.57	271.9
8.96	.375	11.70	30.36	36.33	291.8
10.24	.401	11.69	30.35	38.10	311.8
11.52	.427	11.68	30.34	39.86	331.9
12.80	.453	11.66	30.33	41.63	352.0
14.08	.480	11.65	30.33	43.40	372.1
15.36	.506	11.63	30.32	45.18	392.2
16.64	.532	11.62	30.31	46.95	412.4
17.92	.559	11.60	30.30	48.73	432.7
19.20	.585	11.59	30.29	50.51	453.0
20.48	.612	11.57	30.28	52.29	473.3
21.76	.638	11.55	30.27	54.08	493.7
23.04	.665	11.53	30.26	55.86	514.1
24.32	.691	11.52	30.24	57.65	534.6
25.60	.718	11.50	30.23	59.45	555.1
26.88	.744	11.48	30.22	61.24	575.6
28.16	.771	11.46	30.21	63.04	596.3
29.44	.798	11.44	30.20	64.83	616.8
30.72	.823	11.42	30.19	66.55	636.6
32.00	.847	11.40	30.17	68.17	654.9

HEAT TRANSFER COEFFICIENTS

DISTANCE	ELEMENT	H.T.C.AIR	H.T.C.REF	FINEF	Q.ELE.
.00	1	9.46	70.1	.658	515.79
1.28	2	9.45	70.2	.658	516.19
2.56	3	9.46	70.2	.658	516.58
3.84	4	9.45	70.2	.658	516.97
5.12	5	9.46	70.2	.658	517.42
6.40	6	9.46	70.3	.658	517.91
7.68	7	9.46	70.3	.658	518.41
8.96	8	9.46	70.3	.658	518.93
10.24	9	9.47	70.4	.658	519.49
11.52	10	9.47	70.4	.658	520.07
12.80	11	9.47	70.5	.658	520.68
14.08	12	9.47	70.5	.658	521.31
15.36	13	9.47	70.5	.658	521.97
16.64	14	9.47	70.6	.658	522.65
17.92	15	9.48	70.6	.658	523.35
19.20	16	9.48	70.7	.658	524.08
20.48	17	9.48	70.7	.658	524.83
21.76	18	9.48	70.8	.658	525.59
23.04	19	9.49	70.8	.658	526.38
24.32	20	9.49	70.9	.657	527.18
25.60	21	9.49	70.9	.657	527.99
26.88	22	9.49	71.0	.657	528.82
28.16	23	9.48	69.7	.658	525.41
29.44	24	9.43	64.6	.658	506.98
30.72	25	9.32	56.7	.660	474.48

CONDENSATION RATES AND WALL TEMPERATURES

DISTANCE	ELEMENT	TEMP.WALL	TEMP.FST	COND.RATE
.00	1	21.62	21.62	.114
1.28	2	21.62	21.62	.114
2.56	3	21.61	21.61	.114
3.84	4	21.61	21.61	.115
5.12	5	21.61	21.61	.115
6.40	6	21.60	21.60	.115
7.68	7	21.60	21.60	.115
8.96	8	21.59	21.59	.115
10.24	9	21.58	21.58	.115
11.52	10	21.58	21.58	.116
12.80	11	21.57	21.57	.116
14.08	12	21.57	21.57	.116
15.36	13	21.56	21.56	.116
16.64	14	21.55	21.55	.116
17.92	15	21.54	21.54	.117
19.20	16	21.54	21.54	.117
20.48	17	21.53	21.53	.117
21.76	18	21.52	21.52	.117
23.04	19	21.51	21.51	.117
24.32	20	21.50	21.50	.118
25.60	21	21.50	21.50	.118
26.88	22	21.49	21.49	.118
28.16	23	21.53	21.53	.117
29.44	24	21.73	21.73	.112
30.72	25	22.09	22.09	.102

RESISTANCE VALUES

DISTANCE	ELEMENT	RESA	RESFST	RESR	FST THK	VLOCAL
.00	1	.00716	.00023	.01418	.00448	494.59
1.28	2	.00716	.00023	.01418	.00448	494.59
2.56	3	.00716	.00023	.01417	.00449	494.59
3.84	4	.00716	.00023	.01417	.00450	494.59
5.12	5	.00716	.00023	.01416	.00450	494.58
6.40	6	.00716	.00023	.01415	.00451	494.58
7.68	7	.00715	.00023	.01415	.00452	494.58
8.96	8	.00715	.00023	.01414	.00453	494.57
10.24	9	.00715	.00023	.01413	.00454	494.57
11.52	10	.00715	.00023	.01413	.00455	494.56
12.80	11	.00715	.00023	.01412	.00456	494.56
14.08	12	.00715	.00023	.01411	.00457	494.55
15.36	13	.00715	.00023	.01410	.00458	494.55
16.64	14	.00715	.00024	.01409	.00459	494.54
17.92	15	.00715	.00024	.01408	.00460	494.54
19.20	16	.00714	.00024	.01407	.00462	494.53
20.48	17	.00714	.00024	.01406	.00463	494.53
21.76	18	.00714	.00024	.01406	.00464	494.52
23.04	19	.00714	.00024	.01405	.00465	494.52
24.32	20	.00714	.00024	.01404	.00467	494.51
25.60	21	.00714	.00024	.01403	.00468	494.51
26.88	22	.00714	.00024	.01402	.00470	494.50
28.16	23	.00714	.00024	.01426	.00464	494.54
29.44	24	.00717	.00022	.01539	.00432	494.68
30.72	25	.00724	.00018	.01756	.00379	494.92

THE COIL IS FROSTED

TIME= 1.00

CALCULATED OVERALL, AND AVERAGE RESULTS

PERCENT OF COIL IN 2 PHASE REF.REGION

SATURATED REGION= 100.0 %

AIR FLOW= 6115.2 LBM/HR   REF.FLOW= 294.0LBM/HR

AVG FROST THK IN 2-PHASE REF.REGION.

AVERAGE THICKNESS= .008190 INCHES   ACC.FROST= 2.880 LBM

HEAT TRANSFER RATES (BTU/HR)

QTOTAL=12976.7   QSENSIBLE= 9469.3
QLATENT= 3507.4

CONDENSATION RATE (LBM/HR)

TOTAL CONDENSATION RATE= 2.877

AVG.QTYS.IN THE 2 PHASE REF.REGION

```
LEAVING DRY BULB  = 21.05 F
LEAVING WET BULB  = 21.04 F
WALL TEMPERATURE  = 21.57 F
REL. HUMIDITY     =  99.9 %
LEAVING HUMIDITY  =   .2252E-02 LBWV/LBDA
LATENT ENTHALPY   =  2.39 BTU/LBM
SENSIBLE ENTHALPY=  5.07 BTU/LBM
TOTAL ENTHALPY    =  7.46 BTU/LBM
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AVG.QTYS. OVER LENGTH OF COIL

```
LEAVING DRY BULB  = 21.05 F
LEAVING WET BULB  = 21.04 F
WALL TEMPERATURE  = 21.57 F
REL. HUMIDITY     =  99.9 %
LEAVING HUMIDITY  =   .2252E-02 LBWV/LBDA
LATENT ENTHALPY   =  2.39 BTU/LBM
SENSIBLE ENTHALPY=  5.07 BTU/LBM
TOTAL ENTHALPY    =  7.46 BTU/LBM
```

E N T E R I N G   A I R   T E M P E R A T U R E S

ELEM.	DIST	DR.B.	WT.B.	R.H.	SP.HU.	ENTH	D.PT	S.H.	L.H.	S.V.
1	.00	27.75	26.72	.880	.27E-02	9.6	25.0	6.7	2.89	12.3
2	1.28	27.75	26.72	.880	.27E-02	9.6	25.0	6.7	2.89	12.3
3	2.56	27.75	26.72	.880	.27E-02	9.6	25.0	6.7	2.89	12.3
4	3.84	27.75	26.72	.880	.27E-02	9.6	25.0	6.7	2.89	12.3
5	5.12	27.75	26.72	.880	.27E-02	9.6	25.0	6.7	2.89	12.3
6	6.40	27.75	26.72	.880	.27E-02	9.6	25.0	6.7	2.89	12.3
7	7.68	27.75	26.72	.880	.27E-02	9.6	25.0	6.7	2.89	12.3
8	8.96	27.75	26.72	.880	.27E-02	9.6	25.0	6.7	2.89	12.3
9	10.24	27.75	26.72	.880	.27E-02	9.6	25.0	6.7	2.89	12.3
10	11.52	27.75	26.72	.880	.27E-02	9.6	25.0	6.7	2.89	12.3
11	12.80	27.75	26.72	.880	.27E-02	9.6	25.0	6.7	2.89	12.3
12	14.08	27.75	26.72	.880	.27E-02	9.6	25.0	6.7	2.89	12.3
13	15.36	27.75	26.72	.880	.27E-02	9.6	25.0	6.7	2.89	12.3
14	16.64	27.75	26.72	.880	.27E-02	9.6	25.0	6.7	2.89	12.3
15	17.92	27.75	26.72	.880	.27E-02	9.6	25.0	6.7	2.89	12.3
16	19.20	27.75	26.72	.880	.27E-02	9.6	25.0	6.7	2.89	12.3
17	20.48	27.75	26.72	.880	.27E-02	9.6	25.0	6.7	2.89	12.3
18	21.76	27.75	26.72	.880	.27E-02	9.6	25.0	6.7	2.89	12.3
19	23.04	27.75	26.72	.880	.27E-02	9.6	25.0	6.7	2.89	12.3
20	24.32	27.75	26.72	.880	.27E-02	9.6	25.0	6.7	2.89	12.3
21	25.60	27.75	26.72	.880	.27E-02	9.6	25.0	6.7	2.89	12.3
22	26.88	27.75	26.72	.880	.27E-02	9.6	25.0	6.7	2.89	12.3
23	28.16	27.75	26.72	.880	.27E-02	9.6	25.0	6.7	2.89	12.3
24	29.44	27.75	26.72	.880	.27E-02	9.6	25.0	6.7	2.89	12.3
25	30.72	27.75	26.72	.880	.27E-02	9.6	25.0	6.7	2.89	12.3

LEAVING AIR TEMPERATURES

ELEM.	DIST	DR.B.	WT.B.	R.H.	SP.HU.	ENTH	D.PT	S.H.	L.H.	S.V.
1	.00	21.08	21.08	1.000	.23E-02	7.5	21.1	5.1	2.39	12.2
2	1.28	21.08	21.07	1.000	.23E-02	7.5	21.1	5.1	2.39	12.2
3	2.56	21.07	21.07	1.000	.23E-02	7.5	21.1	5.1	2.39	12.2
4	3.84	21.07	21.06	1.000	.23E-02	7.5	21.1	5.1	2.39	12.2
5	5.12	21.06	21.06	1.000	.23E-02	7.5	21.1	5.1	2.39	12.2
6	6.40	21.06	21.05	1.000	.23E-02	7.5	21.1	5.1	2.39	12.2
7	7.68	21.05	21.05	1.000	.23E-02	7.5	21.0	5.1	2.39	12.2
8	8.96	21.04	21.04	1.000	.23E-02	7.5	21.0	5.1	2.39	12.2
9	10.24	21.04	21.03	1.000	.23E-02	7.5	21.0	5.1	2.39	12.2
10	11.52	21.03	21.03	1.000	.23E-02	7.5	21.0	5.1	2.39	12.2
11	12.80	21.02	21.02	1.000	.23E-02	7.5	21.0	5.1	2.39	12.2
12	14.08	21.02	21.01	1.000	.22E-02	7.5	21.0	5.1	2.39	12.2
13	15.36	21.01	21.00	1.000	.22E-02	7.4	21.0	5.1	2.39	12.2
14	16.64	21.00	21.00	1.000	.22E-02	7.4	21.0	5.1	2.38	12.2
15	17.92	20.99	20.99	1.000	.22E-02	7.4	21.0	5.1	2.38	12.2
16	19.20	20.98	20.98	1.000	.22E-02	7.4	21.0	5.1	2.38	12.2
17	20.48	20.97	20.97	1.000	.22E-02	7.4	21.0	5.1	2.38	12.2
18	21.76	20.96	20.96	1.000	.22E-02	7.4	21.0	5.1	2.38	12.2
19	23.04	20.95	20.95	1.000	.22E-02	7.4	20.9	5.0	2.38	12.2
20	24.32	20.94	20.94	1.000	.22E-02	7.4	20.9	5.0	2.38	12.2
21	25.60	20.93	20.93	1.000	.22E-02	7.4	20.9	5.0	2.38	12.2
22	26.88	20.92	20.92	1.000	.22E-02	7.4	20.9	5.0	2.38	12.2
23	28.16	20.96	20.96	1.000	.22E-02	7.4	21.0	5.1	2.38	12.2
24	29.44	21.18	21.18	1.000	.23E-02	7.5	21.2	5.1	2.41	12.2
25	30.72	21.58	21.58	1.000	.23E-02	7.7	21.6	5.2	2.45	12.2

R E F R I G E R A N T T E M P E R A T U R E S

DISTANCE	QUALITY	TEMP.	PRES.	ENTH.	REF.VEL.
.00	.192	11.77	30.40	24.02	152.6
1.28	.218	11.76	30.40	25.77	172.4
2.56	.244	11.76	30.39	27.53	192.2
3.84	.270	11.75	30.38	29.28	212.1
5.12	.296	11.74	30.38	31.04	231.9
6.40	.322	11.73	30.37	32.80	251.8
7.68	.348	11.72	30.37	34.56	271.8
8.96	.375	11.70	30.36	36.32	291.7
10.24	.401	11.69	30.35	38.08	311.7
11.52	.427	11.68	30.34	39.85	331.7
12.80	.453	11.66	30.33	41.62	351.8
14.08	.479	11.65	30.33	43.39	371.9
15.36	.506	11.63	30.32	45.16	392.0
16.64	.532	11.62	30.31	46.93	412.2
17.92	.558	11.60	30.30	48.70	432.4
19.20	.585	11.59	30.29	50.48	452.6
20.48	.611	11.57	30.28	52.26	472.9
21.76	.637	11.55	30.27	54.04	493.2
23.04	.664	11.53	30.26	55.82	513.6
24.32	.690	11.52	30.24	57.61	534.0
25.60	.717	11.50	30.23	59.39	554.5
26.88	.744	11.48	30.22	61.19	575.0
28.16	.770	11.46	30.21	62.98	595.5
29.44	.797	11.44	30.20	64.76	616.0
30.72	.822	11.42	30.19	66.49	635.8
32.00	.846	11.40	30.17	68.12	654.3

HEAT TRANSFER COEFFICIENTS

DISTANCE	ELEMENT	H.T.C.AIR	H.T.C.REF	FINEF	Q.ELE.
.00	1	9.87	70.1	.651	515.27
1.28	2	9.87	70.1	.651	515.95
2.56	3	9.88	70.2	.651	516.19
3.84	4	9.88	70.2	.651	516.61
5.12	5	9.88	70.2	.651	517.02
6.40	6	9.88	70.2	.651	517.46
7.68	7	9.88	70.3	.651	517.93
8.96	8	9.88	70.3	.651	518.43
10.24	9	9.89	70.3	.651	518.95
11.52	10	9.89	70.4	.651	519.50
12.80	11	9.89	70.4	.651	520.07
14.08	12	9.89	70.4	.651	520.42
15.36	13	9.89	70.5	.651	520.97
16.64	14	9.90	70.5	.651	521.58
17.92	15	9.90	70.5	.651	522.23
19.20	16	9.90	70.6	.651	522.89
20.48	17	9.90	70.6	.651	523.58
21.76	18	9.91	70.7	.651	524.28
23.04	19	9.91	70.7	.651	525.00
24.32	20	9.91	70.8	.651	525.73
25.60	21	9.91	70.8	.651	526.48
26.88	22	9.91	70.9	.651	527.24
28.16	23	9.90	69.8	.651	524.52
29.44	24	9.85	64.9	.652	508.08
30.72	25	9.73	57.1	.654	478.01

CONDENSATION RATES AND WALL TEMPERATURES

DISTANCE	ELEMENT	TEMP.WALL	TEMP.FST	COND.RATE
.00	1	21.61	21.73	.114
1.28	2	21.61	21.73	.114
2.56	3	21.60	21.72	.114
3.84	4	21.60	21.72	.114
5.12	5	21.59	21.71	.115
6.40	6	21.59	21.71	.115
7.68	7	21.58	21.70	.115
8.96	8	21.58	21.70	.115
10.24	9	21.57	21.69	.115
11.52	10	21.57	21.69	.115
12.80	11	21.56	21.68	.116
14.08	12	21.55	21.67	.116
15.36	13	21.54	21.66	.116
16.64	14	21.53	21.66	.116
17.92	15	21.52	21.65	.116
19.20	16	21.52	21.64	.116
20.48	17	21.51	21.63	.117
21.76	18	21.50	21.62	.117
23.04	19	21.49	21.62	.117
24.32	20	21.48	21.61	.117
25.60	21	21.47	21.60	.118
26.88	22	21.46	21.59	.118
28.16	23	21.51	21.63	.117
29.44	24	21.72	21.83	.112
30.72	25	22.09	22.18	.102

RESISTANCE VALUES

DISTANCE	ELEMENT	RESA	RESFST	RESR	FST THK	VLOCAL
.00	1	.00696	.00037	.01419	.00809	537.84
1.28	2	.00696	.00037	.01418	.00810	537.85
2.56	3	.00696	.00037	.01418	.00811	537.84
3.84	4	.00696	.00037	.01417	.00812	537.84
5.12	5	.00696	.00037	.01417	.00814	537.83
6.40	6	.00696	.00037	.01416	.00815	537.83
7.68	7	.00696	.00038	.01415	.00817	537.83
8.96	8	.00696	.00038	.01415	.00818	537.82
10.24	9	.00696	.00038	.01414	.00820	537.82
11.52	10	.00696	.00038	.01413	.00822	537.81
12.80	11	.00696	.00038	.01413	.00824	537.81
14.08	12	.00695	.00038	.01412	.00825	537.80
15.36	13	.00695	.00038	.01411	.00827	537.79
16.64	14	.00695	.00038	.01411	.00829	537.79
17.92	15	.00695	.00038	.01410	.00831	537.78
19.20	16	.00695	.00039	.01409	.00833	537.77
20.48	17	.00695	.00039	.01408	.00836	537.77
21.76	18	.00695	.00039	.01407	.00838	537.76
23.04	19	.00695	.00039	.01406	.00840	537.75
24.32	20	.00695	.00039	.01406	.00843	537.75
25.60	21	.00694	.00039	.01405	.00845	537.74
26.88	22	.00694	.00039	.01404	.00847	537.73
28.16	23	.00695	.00039	.01426	.00838	537.77
29.44	24	.00698	.00035	.01533	.00783	537.93
30.72	25	.00704	.00030	.01742	.00688	538.22

THREE REGION MODEL OUTPUT

INPUT PARAMETERS

GEOMETRIC DATA. ALL DATA IN UNITS OF FEET.

FACE A.= 4.000 WIDTH= 8.000 HEIGHT= .500 DEPTH= .750

FPI= 8.0 ROWS H.= 4.0 CKTS= 6.0

ROW HEIGHT= .125 ROW DEPTH= .125 DIA.O.= .0422 DIA.I.= .0417

FIN THK.= .00050 FIN SPAC.= .0104 FIN H.= .0414 FINS= 3072.0

ROWS= 6.0 L. CKT= 32.0 KFIN=128.00 BTU/HR*FT*F

ENTERING AIR AND REFRIGERANT STATES

TA1=27.75 F RH= .880 PRA= .718 BTU/LB*F KA= .0137 BTU/HR*F*FT

MFA= 6115.2 LB/HR MUA= .0411 LB/FT*HR HFGW=1219.0 BTU/LB

TR=11.77 F HR1= 24.02 BTU/LBM XIN= .19 PR1=30.40 PSI

SPECIFIED LVG.REF.QUALITY= .839E+00

L E A V I N G   C O N D I T I O N S.

COIL IS FROSTED

TIME= .50 HRS
AIR MASS FLOW = 6115.2 LBDA/HR
FACE VELOCITY = 314.5 FPM
REFR. FLOW = 294.0 LBM/HR

H E A T   T R A N S F E R E D   (BTU/HR)

Q2PH=12826.3   QSUP= .0   QTOT=12826.3
QLAT= 3455.9

C O N D E N S A T I O N (LBWV/HR)

CONDENSATION= 2.836 LBWV/HR

S A T U R A T E D   R E G I O N.

PERCENT OF COIL IN 2-PHASE REGION=100.0 %

FST.THK.= .0044 IN   MASS FROST= 1.418 LBM

TA2= 21.13 F   W2= .226E-02 LBWV/LBDA   RH2= 99.9 %
TFR= 21.66 F   TWALL= 21.66 F   ENTHALPY= 7.49 BTU/LBM

S U P E R H E A T E D   R E G I O N.

TA2= .00 F   W2= .000E+00 LBWV/LBDA   RH2= .0 %
TWALL= .00 F   ENTHALPY= .00 BTU/LBM

M I X E D   C O N D I T I O N S.

TA2= 21.13 F   W2= .226E-02 LBWV/LBDA   RH2= 99.9 %
TWALL= 21.66 F   ENTHALPY= 7.49 BTU/LBM

R E F R I G E R A N T   L V G.

TR2=11.77 F   SUP.HEAT= .00 F   X2= .839

H E A T   T R A N S F E R   P A R A M E T E R S

S A T U R A T E D   R E G I O N.

HTA= 9.45   HTR= 69.32   FIN EFF= .658

RESA=.0005742   RESR=.0011841   RESFST=.0000090

S U P E R H E A T E D   R E G I O N.

HTA= .00   HTR= .00   FIN EFF= .000

RESA=.0000000   RESR=.0000000   RESFST=.0000000

L E A V I N G   C O N D I T I O N S.

COIL IS FROSTED

TIME= 1.00 HRS
AIR MASS FLOW = 6115.2 LBDA/HR
FACE VELOCITY = 314.5 FPM
REFR. FLOW = 294.0 LBM/HR

H E A T   T R A N S F E R E D   (BTU/HR)

Q2PH=12809.7   QSUP= .0   QTOT=12809.7
QLAT= 3450.4

C O N D E N S A T I O N (LBWV/HR)

CONDENSATION= 2.831 LBWV/HR

S A T U R A T E D   R E G I O N.

PERCENT OF COIL IN 2-PHASE REGION=100.0 %

FST.THK.= .0080 IN   MASS FROST= 2.833 LBM

TA2= 21.14 F   W2= .226E-02 LBWV/LBDA   RH2= 99.9 %
TFR= 21.76 F   TWALL= 21.64 F   ENTHALPY= 7.49 BTU/LBM

S U P E R H E A T E D   R E G I O N.

TA2= .00 F   W2= .000E+00 LBWV/LBDA   RH2= .0 %
TWALL= .00 F   ENTHALPY= .00 BTU/LBM

M I X E D   C O N D I T I O N S.

TA2= 21.14 F   W2= .226E-02 LBWV/LBDA   RH2= 99.9 %
TWALL= 21.64 F   ENTHALPY= 7.49 BTU/LBM

R E F R I G E R A N T   L V G.

TR2=11.77 F   SUP.HEAT= .00 F   X2= .838

H E A T   T R A N S F E R   P A R A M E T E R S

S A T U R A T E D   R E G I O N.

HTA= 9.85   HTR= 69.29   FIN EFF= .652

RESA=.0005589   RESR=.0011834   RESFST=.0000146

S U P E R H E A T E D   R E G I O N.

HTA= .00   HTR= .00   FIN EFF= .000

RESA=.0000000   RESR=.0000000   RESFST=.0000000

L E A V I N G   C O N D I T I O N S.

COIL IS FROSTED

```
             TIME= 1.50 HRS
AIR MASS FLOW =  6115.2 LBDA/HR
FACE VELOCITY =   314.5 FPM
REFR. FLOW    =   294.0 LBM/HR
```

H E A T   T R A N S F E R E D   (BTU/HR)

```
Q2PH=12814.6   QSUP=      .0   QTOT=12814.6
QLAT= 3452.3
```

C O N D E N S A T I O N (LBWV/HR)

```
 CONDENSATION= 2.833 LBWV/HR
```

S A T U R A T E D   R E G I O N.

```
PERCENT OF COIL IN 2-PHASE REGION=100.0 %

FST.THK.= .0110 IN   MASS FROST= 4.249 LBM

TA2= 21.14 F   W2=  .226E-02 LBWV/LBDA   RH2= 99.9 %
TFR= 21.83 F     TWALL= 21.64 F    ENTHALPY= 7.49 BTU/LBM
```

S U P E R H E A T E D   R E G I O N.

```
TA2=    .00 F   W2=  .000E+00 LBWV/LBDA   RH2=    .0 %
TWALL=    .00 F    ENTHALPY=   .00 BTU/LBM
```

M I X E D   C O N D I T I O N S.

```
TA2= 21.14 F   W2=  .226E-02 LBWV/LBDA   RH2= 99.9 %
TWALL= 21.64 F    ENTHALPY= 7.49 BTU/LBM
```

R E F R I G E R A N T   L V G.

```
TR2=11.77 F   SUP.HEAT=   .00 F   X2= .838
```

H E A T   T R A N S F E R   P A R A M E T E R S

S A T U R A T E D   R E G I O N.

```
HTA=10.21     HTR= 69.30    FIN EFF= .646

RESA=.0005462   RESR=.0011829   RESFST=.0000184
```

S U P E R H E A T E D   R E G I O N.

```
HTA=   .00     HTR=    .00    FIN EFF= .000

RESA=.0000000   RESR=.0000000   RESFST=.0000000
```

L E A V I N G   C O N D I T I O N S.

COIL IS FROSTED

```
            TIME= 2.00 HRS
AIR MASS FLOW =  6115.2 LBDA/HR
FACE VELOCITY =   314.5 FPM
REFR. FLOW    =   294.0 LBM/HR
```

H E A T   T R A N S F E R E D   (BTU/HR)

```
Q2PH=12846.2   QSUP=      .0   QTOT=12846.2
QLAT= 3463.4
```

C O N D E N S A T I O N   (LBWV/HR)

```
CONDENSATION= 2.842 LBWV/HR
```

S A T U R A T E D   R E G I O N.

```
PERCENT OF COIL IN 2-PHASE REGION=100.0 %

FST.THK.= .0137 IN   MASS FROST= 5.671 LBM

TA2= 21.11 F   W2=  .226E-02 LBWV/LBDA   RH2= 99.9 %
TFR= 21.89 F    TWALL= 21.65 F   ENTHALPY= 7.48 BTU/LBM
```

S U P E R H E A T E D   R E G I O N.

```
TA2=    .00 F   W2=  .000E+00 LBWV/LBDA   RH2=   .0 %
TWALL=    .00 F    ENTHALPY=  .00 BTU/LBM
```

M I X E D   C O N D I T I O N S.

```
TA2= 21.11 F   W2=  .226E-02 LBWV/LBDA   RH2= 99.9 %
TWALL= 21.65 F    ENTHALPY= 7.48 BTU/LBM
```

R E F R I G E R A N T   L V G.

```
TR2=11.77 F   SUP.HEAT=  .00 F   X2= .840
```

H E A T   T R A N S F E R   P A R A M E T E R S

S A T U R A T E D   R E G I O N.

```
HTA=10.60    HTR= 69.35   FIN EFF= .640

RESA=.0005334   RESR=.0011827   RESFST=.0000211
```

S U P E R H E A T E D   R E G I O N.

```
HTA=  .00    HTR=    .00   FIN EFF= .000

RESA=.0000000   RESR=.0000000   RESFST=.0000000
```

```
                 L E A V I N G   C O N D I T I O N S.
                 ====================================

                         COIL IS FROSTED
                         ---------------

                        TIME= 3.00 HRS
                 AIR MASS FLOW =  6115.2 LBDA/HR
                 FACE VELOCITY =   314.5 FPM
                 REFR. FLOW    =  294.0 LBM/HR

                H E A T  T R A N S F E R E D  (BTU/HR)
                --------------------------------------
              Q2PH=12877.8   QSUP=      .0   QTOT=12877.8
              QLAT= 3474.9

             C O N D E N S A T I O N (LBWV/HR)
             ---------------------------------
              CONDENSATION= 2.852 LBWV/HR

             S A T U R A T E D  R E G I O N.
             -------------------------------
           PERCENT OF COIL IN 2-PHASE REGION=100.0 %

           FST.THK.= .0183 IN   MASS FROST= 8.523 LBM

           TA2= 21.10 F   W2=  .226E-02 LBWV/LBDA   RH2= 99.9 %
           TFR= 21.94 F    TWALL= 21.67 F   ENTHALPY= 7.48 BTU/LBM

             S U P E R H E A T E D  R E G I O N.
             -----------------------------------
           TA2=    .00 F   W2=  .000E+00 LBWV/LBDA   RH2=    .0 %
           TWALL=    .00 F   ENTHALPY=  .00 BTU/LBM

             M I X E D  C O N D I T I O N S.
             -------------------------------
           TA2= 21.10 F   W2=  .226E-02 LBWV/LBDA   RH2= 99.9 %
           TWALL= 21.67 F   ENTHALPY= 7.48 BTU/LBM

             R E F R I G E R A N T  L V G.
             -----------------------------
           TR2=11.77 F   SUP.HEAT=   .00 F   X2= .841
           H E A T  T R A N S F E R  P A R A M E T E R S
          ==============================================

             S A T U R A T E D  R E G I O N.
             -------------------------------
           HTA=10.95    HTR= 69.40   FIN EFF= .635

           RESA=.0005223   RESR=.0011823   RESFST=.0000249

             S U P E R H E A T E D  R E G I O N.
             -----------------------------------
           HTA=   .00    HTR=    .00   FIN EFF= .000

           RESA=.0000000   RESR=.0000000   RESFST=.0000000
```

L E A V I N G   C O N D I T I O N S.

COIL IS FROSTED

TIME= 4.00 HRS
AIR MASS FLOW = 6115.2 LBDA/HR
FACE VELOCITY = 314.5 FPM
REFR. FLOW = 294.0 LBM/HR

H E A T   T R A N S F E R E D (BTU/HR)

Q2PH=12955.3   QSUP= .0   QTOT=12955.3
QLAT= 3503.4

C O N D E N S A T I O N (LBWV/HR)

CONDENSATION= 2.875 LBWV/HR

S A T U R A T E D   R E G I O N.

PERCENT OF COIL IN 2-PHASE REGION=100.0 %

FST.THK.= .0219 IN   MASS FROST=11.398 LBM

TA2= 21.06 F   W2= .225E-02 LBWV/LBDA   RH2= 99.9 %
TFR= 22.02 F   TWALL= 21.70 F   ENTHALPY= 7.46 BTU/LBM

S U P E R H E A T E D   R E G I O N.

TA2= .00 F   W2= .000E+00 LBWV/LBDA   RH2= .0 %
TWALL= .00 F   ENTHALPY= .00 BTU/LBM

M I X E D   C O N D I T I O N S.

TA2= 21.06 F   W2= .225E-02 LBWV/LBDA   RH2= 99.9 %
TWALL= 21.70 F   ENTHALPY= 7.46 BTU/LBM

R E F R I G E R A N T   L V G.

TR2=11.77 F   SUP.HEAT= .00 F   X2= .845

H E A T   T R A N S F E R   P A R A M E T E R S

S A T U R A T E D   R E G I O N.

HTA=11.66   HTR= 69.53   FIN EFF= .625

RESA=.0005018   RESR=.0011822   RESFST=.0000267

S U P E R H E A T E D   R E G I O N.

HTA= .00   HTR= .00   FIN EFF= .000

RESA=.0000000   RESR=.0000000   RESFST=.0000000

LEAVING CONDITIONS.

COIL IS FROSTED

TIME= 5.00 HRS
AIR MASS FLOW = 6115.2 LBDA/HR
FACE VELOCITY = 314.5 FPM
REFR. FLOW = 301.9 LBM/HR

HEAT TRANSFERED (BTU/HR)

Q2PH=13184.9 QSUP= .0 QTOT=13184.9
QLAT= 3588.0

CONDENSATION (LBWV/HR)

CONDENSATION= 2.946 LBWV/HR

SATURATED REGION.

PERCENT OF COIL IN 2-PHASE REGION=100.0 %

FST.THK.= .0254 IN MASS FROST=14.343 LBM

TA2= 20.96 F W2= .224E-02 LBWV/LBDA RH2= 99.9 %
TFR= 22.04 F TWALL= 21.69 F ENTHALPY= 7.43 BTU/LBM

SUPERHEATED REGION.

TA2= .00 F W2= .000E+00 LBWV/LBDA RH2= .0 %
TWALL= .00 F ENTHALPY= .00 BTU/LBM

MIXED CONDITIONS.

TA2= 20.96 F W2= .224E-02 LBWV/LBDA RH2= 99.9 %
TWALL= 21.69 F ENTHALPY= 7.43 BTU/LBM

REFRIGERANT LVG.

TR2=11.77 F SUP.HEAT= .00 F X2= .840

HEAT TRANSFER PARAMETERS

SATURATED REGION.

HTA=12.36 HTR= 71.22 FIN EFF= .615

RESA=.0004830 RESR=.0011487 RESFST=.0000280

SUPERHEATED REGION.

HTA= .00 HTR= .00 FIN EFF= .000

RESA=.0000000 RESR=.0000000 RESFST=.0000000

L E A V I N G   C O N D I T I O N S.

COIL IS FROSTED

```
          TIME= 6.00 HRS
AIR MASS FLOW =  6115.2 LBDA/HR
FACE VELOCITY =   314.5 FPM
REFR. FLOW    =   301.9 LBM/HR
```

H E A T   T R A N S F E R E D   (BTU/HR)

```
Q2PH=13280.4    QSUP=        .0    QTOT=13280.4
QLAT= 3621.8
```

C O N D E N S A T I O N (LBWV/HR)

```
CONDENSATION= 2.973 LBWV/HR
```

S A T U R A T E D   R E G I O N.

```
PERCENT OF COIL IN 2-PHASE REGION=100.0 %

FST.THK.= .0280 IN   MASS FROST=17.317 LBM

TA2= 20.92 F   W2=  .224E-02 LBWV/LBDA   RH2= 99.9 %
TFR= 22.10 F     TWALL= 21.73 F    ENTHALPY= 7.41 BTU/LBM
```

S U P E R H E A T E D   R E G I O N.

```
TA2=    .00 F   W2=  .000E+00 LBWV/LBDA   RH2=    .0 %
TWALL=    .00 F    ENTHALPY=   .00 BTU/LBM
```

M I X E D   C O N D I T I O N S.

```
TA2= 20.92 F   W2=  .224E-02 LBWV/LBDA   RH2= 99.9 %
TWALL= 21.73 F    ENTHALPY= 7.41 BTU/LBM
```

R E F R I G E R A N T   L V G.

```
TR2=11.77 F    SUP.HEAT=  .00 F    X2= .844
```

H E A T   T R A N S F E R   P A R A M E T E R S

S A T U R A T E D   R E G I O N.

```
HTA=13.12      HTR= 71.36    FIN EFF= .605

RESA=.0004649    RESR=.0011513    RESFST=.0000281
```

S U P E R H E A T E D   R E G I O N.

```
HTA=   .00     HTR=    .00    FIN EFF= .000

RESA=.0000000    RESR=.0000000    RESFST=.0000000
```

L E A V I N G   C O N D I T I O N S.

COIL IS FROSTED

TIME= 7.00 HRS
AIR MASS FLOW = 5695.6 LBDA/HR
FACE VELOCITY = 292.9 FPM
REFR. FLOW = 296.4 LBM/HR

H E A T   T R A N S F E R E D (BTU/HR)

Q2PH=12934.8 QSUP= .0 QTOT=12934.8
QLAT= 3583.1

C O N D E N S A T I O N (LBWV/HR)

CONDENSATION= 2.940 LBWV/HR

S A T U R A T E D   R E G I O N.

PERCENT OF COIL IN 2-PHASE REGION=100.0 %

FST.THK.= .0311 IN MASS FROST=20.257 LBM

TA2= 20.63 F W2= .221E-02 LBWV/LBDA RH2= 99.9 %
TFR= 21.99 F TWALL= 21.63 F ENTHALPY= 7.31 BTU/LBM

S U P E R H E A T E D   R E G I O N.

TA2= .00 F W2= .000E+00 LBWV/LBDA RH2= .0 %
TWALL= .00 F ENTHALPY= .00 BTU/LBM

M I X E D   C O N D I T I O N S.

TA2= 20.63 F W2= .221E-02 LBWV/LBDA RH2= 99.9 %
TWALL= 21.63 F ENTHALPY= 7.31 BTU/LBM

R E F R I G E R A N T   L V G.

TR2=11.77 F SUP.HEAT= .00 F X2= .839

H E A T   T R A N S F E R   P A R A M E T E R S

S A T U R A T E D   R E G I O N.

HTA=13.27 HTR= 69.89 FIN EFF= .603

RESA=.0004631 RESR=.0011682 RESFST=.0000292

S U P E R H E A T E D   R E G I O N.

HTA= .00 HTR= .00 FIN EFF= .000

RESA=.0000000 RESR=.0000000 RESFST=.0000000

L E A V I N G   C O N D I T I O N S.

COIL IS FROSTED

TIME= 8.00 HRS
AIR MASS FLOW = 4962.6 LBDA/HR
FACE VELOCITY = 255.2 FPM
REFR. FLOW = 276.3 LBM/HR

H E A T   T R A N S F E R E D   (BTU/HR)

Q2PH=12095.2   QSUP= .0   QTOT=12095.2
QLAT= 3418.1

C O N D E N S A T I O N (LBWV/HR)

CONDENSATION= 2.805 LBWV/HR

S A T U R A T E D   R E G I O N.

PERCENT OF COIL IN 2-PHASE REGION=100.0 %

FST.THK.= .0335 IN   MASS FROST=23.055 LBM

TA2= 20.16 F   W2= .216E-02 LBWV/LBDA   RH2= 99.9 %
TFR= 21.85 F   TWALL= 21.49 F   ENTHALPY= 7.15 BTU/LBM

S U P E R H E A T E D   R E G I O N.

TA2= .00 F   W2= .000E+00 LBWV/LBDA   RH2= .0 %
TWALL= .00 F   ENTHALPY= .00 BTU/LBM

M I X E D   C O N D I T I O N S.

TA2= 20.16 F   W2= .216E-02 LBWV/LBDA   RH2= 99.9 %
TWALL= 21.49 F   ENTHALPY= 7.15 BTU/LBM

R E F R I G E R A N T   L V G.

TR2=11.77 F   SUP.HEAT= .00 F   X2= .841

H E A T   T R A N S F E R   P A R A M E T E R S

S A T U R A T E D   R E G I O N.

HTA=13.54   HTR= 65.22   FIN EFF= .599

RESA=.0004597   RESR=.0012552   RESFST=.0000296

S U P E R H E A T E D   R E G I O N.

HTA= .00   HTR= .00   FIN EFF= .000

RESA=.0000000   RESR=.0000000   RESFST=.0000000

L E A V I N G   C O N D I T I O N S.

COIL IS FROSTED

TIME= 9.00 HRS,
AIR MASS FLOW = 4317.0 LBDA/HR
FACE VELOCITY = 222.0 FPM
REFR. FLOW = 256.9 LBM/HR

H E A T   T R A N S F E R E D (BTU/HR)

Q2PH=11208.3   QSUP= .0   QTOT=11208.3
QLAT= 3217.6

C O N D E N S A T I O N (LBWV/HR)

CONDENSATION= 2.640 LBWV/HR

S A T U R A T E D   R E G I O N.

PERCENT OF COIL IN 2-PHASE REGION=100.0 %

FST.THK.= .0362 IN   MASS FROST=25.740 LBM

TA2= 19.71 F   W2= .211E-02 LBWV/LBDA   RH2=100.0 %
TFR= 21.65 F   TWALL= 21.31 F   ENTHALPY= 6.99 BTU/LBM

S U P E R H E A T E D   R E G I O N.

TA2= .00 F   W2= .000E+00 LBWV/LBDA   RH2= .0 %
TWALL= .00 F   ENTHALPY= .00 BTU/LBM

M I X E D   C O N D I T I O N S.

TA2= 19.71 F   W2= .211E-02 LBWV/LBDA   RH2=100.0 %
TWALL= 21.31 F   ENTHALPY= 6.99 BTU/LBM

R E F R I G E R A N T   L V G.

TR2=11.77 F   SUP.HEAT= .00 F   X2= .839

H E A T   T R A N S F E R   P A R A M E T E R S

S A T U R A T E D   R E G I O N.

HTA=13.26   HTR= 60.58   FIN EFF= .603

RESA=.0004687   RESR=.0013504   RESFST=.0000307

S U P E R H E A T E D   R E G I O N.

HTA= .00   HTR= .00   FIN EFF= .000

RESA=.0000000   RESR=.0000000   RESFST=.0000000

LEAVING CONDITIONS.

COIL IS FROSTED

TIME=10.00 HRS
AIR MASS FLOW = 3761.3 LBDA/HR
FACE VELOCITY = 193.4 FPM
REFR. FLOW = 236.6 LBM/HR

HEAT TRANSFERED (BTU/HR)

Q2PH=10323.4 QSUP= .0 QTOT=10323.4
QLAT= 3000.4

CONDENSATION (LBWV/HR)

CONDENSATION= 2.462 LBWV/HR

SATURATED REGION.

PERCENT OF COIL IN 2-PHASE REGION=100.0 %

FST.THK.= .0385 IN MASS FROST=28.266 LBM

TA2= 19.28 F W2= .207E-02 LBWV/LBDA RH2=100.0 %
TFR= 21.47 F TWALL= 21.15 F ENTHALPY= 6.84 BTU/LBM

SUPERHEATED REGION.

TA2= .00 F W2= .000E+00 LBWV/LBDA RH2= .0 %
TWALL= .00 F ENTHALPY= .00 BTU/LBM

MIXED CONDITIONS.

TA2= 19.28 F W2= .207E-02 LBWV/LBDA RH2=100.0 %
TWALL= 21.15 F ENTHALPY= 6.84 BTU/LBM

REFRIGERANT LVG.

TR2=11.77 F SUP.HEAT= .00 F X2= .839

HEAT TRANSFER PARAMETERS

SATURATED REGION.

HTA=13.00 HTR= 55.80 FIN EFF= .606

RESA=.0004773 RESR=.0014642 RESFST=.0000316

SUPERHEATED REGION.

HTA= .00 HTR= .00 FIN EFF= .000

RESA=.0000000 RESR=.0000000 RESFST=.0000000

LEAVING CONDITIONS.

COIL IS FROSTED

TIME=11.00 HRS
AIR MASS FLOW = 3281.0 LBDA/HR
FACE VELOCITY = 168.7 FPM
REFR. FLOW = 217.2 LBM/HR

HEAT TRANSFERED (BTU/HR)

Q2PH= 9474.8 QSUP= .0 QTOT= 9474.8
QLAT= 2779.6

CONDENSATION (LBWV/HR)

CONDENSATION= 2.283 LBWV/HR

SATURATED REGION.

PERCENT OF COIL IN 2-PHASE REGION=100.0 %

FST.THK.= .0406 IN MASS FROST=30.618 LBM

TA2= 18.88 F W2= .203E-02 LBWV/LBDA RH2= 99.9 %
TFR= 21.29 F TWALL= 20.99 F ENTHALPY= 6.70 BTU/LBM

SUPERHEATED REGION.

TA2= .00 F W2= .000E+00 LBWV/LBDA RH2= .0 %
TWALL= .00 F ENTHALPY= .00 BTU/LBM

MIXED CONDITIONS.

TA2= 18.88 F W2= .203E-02 LBWV/LBDA RH2= 99.9 %
TWALL= 20.99 F ENTHALPY= 6.70 BTU/LBM

REFRIGERANT LVG.

TR2=11.77 F SUP.HEAT= .00 F X2= .839

HEAT TRANSFER PARAMETERS

SATURATED REGION.

HTA=12.78 HTR= 51.22 FIN EFF= .609

RESA=.0004849 RESR=.0015939 RESFST=.0000322

SUPERHEATED REGION.

HTA= .00 HTR= .00 FIN EFF= .000

RESA=.0000000 RESR=.0000000 RESFST=.0000000

L E A V I N G   C O N D I T I O N S.

COIL IS FROSTED

TIME=12.00 HRS
AIR MASS FLOW = 2897.2 LBDA/HR
FACE VELOCITY = 149.0 FPM
REFR. FLOW = 200.3 LBM/HR

H E A T   T R A N S F E R E D   (BTU/HR)

Q2PH= 8739.1   QSUP= .0   QTOT= 8739.1
QLAT= 2585.4

C O N D E N S A T I O N (LBWV/HR)

CONDENSATION= 2.121 LBWV/HR

S A T U R A T E D   R E G I O N.

PERCENT OF COIL IN 2-PHASE REGION=100.0 %

FST.THK.= .0424 IN   MASS FROST=32.786 LBM

TA2= 18.50 F   W2= .199E-02 LBWV/LBDA   RH2= 99.9 %
TFR= 21.13 F   TWALL= 20.85 F   ENTHALPY= 6.57 BTU/LBM

S U P E R H E A T E D   R E G I O N.

TA2= .00 F   W2= .000E+00 LBWV/LBDA   RH2= .0 %
TWALL= .00 F   ENTHALPY= .00 BTU/LBM

M I X E D   C O N D I T I O N S.

TA2= 18.50 F   W2= .199E-02 LBWV/LBDA   RH2= 99.9 %
TWALL= 20.85 F   ENTHALPY= 6.57 BTU/LBM

R E F R I G E R A N T   L V G.

TR2=11.77 F   SUP.HEAT= .00 F   X2= .839

H E A T   T R A N S F E R   P A R A M E T E R S

S A T U R A T E D   R E G I O N.

HTA=12.71   HTR= 47.24   FIN EFF= .610

RESA=.0004888   RESR=.0017269   RESFST=.0000326

S U P E R H E A T E D   R E G I O N.

HTA= .00   HTR= .00   FIN EFF= .000

RESA=.0000000   RESR=.0000000   RESFST=.0000000

L E A V I N G   C O N D I T I O N S.

COIL IS FROSTED

TIME=13.00 HRS
AIR MASS FLOW = 2545.8 LBDA/HR
FACE VELOCITY = 130.9 FPM
REFR. FLOW = 185.6 LBM/HR

H E A T   T R A N S F E R E D   (BTU/HR)

Q2PH= 8039.9   QSUP= .0   QTOT= 8039.9
QLAT= 2395.2

C O N D E N S A T I O N (LBWV/HR)

CONDENSATION= 1.965 LBWV/HR

S A T U R A T E D   R E G I O N.

PERCENT OF COIL IN 2-PHASE REGION=100.0 %

FST.THK.= .0440 IN   MASS FROST=34.756 LBM

TA2= 18.08 F   W2= .195E-02 LBWV/LBDA   RH2= 99.9 %
TFR= 20.95 F   TWALL= 20.68 F   ENTHALPY= 6.42 BTU/LBM

S U P E R H E A T E D   R E G I O N.

TA2= .00 F   W2= .000E+00 LBWV/LBDA   RH2= .0 %
TWALL= .00 F   ENTHALPY= .00 BTU/LBM

M I X E D   C O N D I T I O N S,

TA2= 18.08 F   W2= .195E-02 LBWV/LBDA   RH2= 99.9 %
TWALL= 20.68 F   ENTHALPY= 6.42 BTU/LBM

R E F R I G E R A N T   L V G.

TR2=11.77 F   SUP.HEAT= .00 F   X2= .834

H E A T   T R A N S F E R   P A R A M E T E R S

S A T U R A T E D   R E G I O N.

HTA=12.71   HTR= 43.66   FIN EFF= .610

RESA=.0004903   RESR=.0018637   RESFST=.0000328

S U P E R H E A T E D   R E G I O N.

HTA= .00   HTR= .00   FIN EFF= .000

RESA=.0000000   RESR=.0000000   RESFST=.0000000

APPENDIX 2

Outline of Finite Element and Three Region Model Programs

Outline of Finite Element Model Program

The coil is divided into a number of geometrically fixed elements. Each element is analyzed separately, and the conditions of the air stream leaving and entering each element, and the refrigerant at each node are stored in arrays. No assumptions regarding the state of the refrigerant in a particular element is made, except that the entering refrigerant condition, i.e. at node 1, is saturated (two-phase).

1 The fluid inlet conditions are specified.

Air side: T_a, φ, $\dot{m}_a$ or ΔP_{int} or V_{fc}

Refrigerant side: h_{ri} or x_i, P_{sat} or T_{sat}, ΔT_{sup} or x_{out} or m_r

2 Coil geometry is specified.
The area calculations are carried out:

$$A_o, A_t, A_f, A_i, A_{fc}, A_{min}$$

3 Initially the coil is assumed dry, and the parameter C, slope of the process line, is set to 0 in the first element.

4 Variables are initialized.

$$G_{max} = m_a / A_{min}$$

$$JFLAG = 0$$

$$KFLAG = 0$$

$$T_{a2}(1) = T_{a1}(1) - 2.0$$

$$T_{av}(1) = (T_{a1}(1) + T_{a2}(1))/2.0$$

$$A_{min}(j) = A_{min} / NELEM$$

$$m_a(j) = G_{max} \cdot A_{min}(j)$$

$$\Delta l = LENGTH / NELEM$$

$$L(j) = L(j-1) + \Delta l$$

$$\Delta x/\Delta l = (x(2) - x(1)) / \Delta l$$

$$q(j) = m_r \cdot (i_{r,g} - i_r(1)) / NELEM$$

5 Analyze each element in a circuit.
Do j = 1 to NELEM

6 Calculate transport properties of the air stream based upon T_{av}.

Set, $T_{av,est} = T_{av}$.

7 Calculate slope of process line, C.

$$C = \frac{w_{av} - w_{s,m}}{T_{av} - T_{s,m}}$$

If C < , set C = 0.

8 From the inlet air condition and the value of C, the air side heat transfer coefficient and the fin effectiveness in element j can be determined:

$$h_a \ , \ \phi_o$$

9 Using m_r, and $\Delta x/\Delta l$, the refrigerant side heat transfer coefficient is determined:

$$h_r$$

10 The thermal resistance terms in an element can now be calculated:

refrigerant side,

$$Res_r = \frac{1}{h_r \cdot A_{i,el} \cdot n}$$

air side,

$$Res_a = \frac{1}{h_a \cdot \left[1 + \frac{i_w \cdot C}{Le \cdot C_{p,da}}\right] \cdot A_{o,el} \cdot \phi_o \cdot n}$$

frost layer,

$$Res_{fsl} = \frac{1}{k_{fsl} \cdot A_{o,el} \cdot n}$$

Fouling (air side and refrigerant side) and contact resistances are accounted for:

$$Res_{f,a},\ Res_{f,r},\ Res_c$$

$$SUMR = \sum_j Res_j$$

thus,

$$q(j) = \frac{(T_{av}(j) - T_r(j))}{SUMR}$$

$$= \frac{\Delta T_{lm}(j)}{SUMR}$$

11 The refrigerant enthalpy at the leaving node (node j+1) is determined:

$$i_r(j+1) = i_r(j) + q(j) / (m_r \cdot n)$$

The quality at node (j+1) is compared to the value calculated at the previous iteration. If they are not equal within acceptable tolerance then:

i) calculate $\Delta x/\Delta l$.

ii) calculate h_r , and Res_r.

iii) determine, $q = \Delta T_{lm} / SUMR$

iv) Calculate x_2. Does it compare to previous value ?

If yes, then go onto calculate surface temperatures; goto 12

If no, go back to i .

12 The surface temperatures are calculated. If the surface temperature of the coil in the element under consideration is below the dew point of the air stream the air is dehumidified. The mean surface temperature in element j is given by:

$$T_{s,m}(j) = T_{av}(j) - q(j) \cdot Res_{a,s,m}$$

$$Res_{a,s,m} = \frac{1}{h_a \cdot A_{o,el} \cdot n}$$

13 If JFLAG = 0, (the dew point of the air stream is compared to $T_{s,m}$).

```
If T_s,m < T_d.pt.   (air is dehumidified)
     JFLAG = 1    (condensate on coil surface)
     If  T_s < 32
     JFLAG = 2    (frosted coil surface)
     end if
     goto 7
   end if

   If |T_s,m - T_s,m,est| < TOLER
     goto 14
   end if
   T_s,m,est = T_s,m
   goto 7
```

14 Calculate enthalpy of air stream leaving coil in element j.

$$i_{a2}(j) = i_{a1}(j) - q(j) / m_a(j)$$

The leaving air stream enthalpy, i_{a2} together with the assumed process line (straight with slope C) between inlet air and coil surface condition in element j is used to determine the state of the leaving air stream.

15 $T_{a1}(j)$, $T_{a2}(j)$, and $T_s(j)$ are used to determine the log mean temperature difference (ΔT_{lm}). The average (log mean) temperature of air in element j as it traverses the coil is therefore:

$$T_{av}(j) = T_s(j) + \Delta T_{lm}(j)$$

where,

$$\Delta T_{lm}^{*}(j) = \frac{T_{a1}(j) - T_{a2}(j)}{\ln\left[\dfrac{T_{a1}(j) - T_{r}(j)}{T_{a2}(j) - T_{r}(j)}\right]}$$

The analysis of an element is completed when convergence on the log mean average temperature in that element is obtained.

Is $|T_{av}(j) - T_{av,est}(j)| <$ TOLERANCE ?

NO : set $T_{av} = 0.75\ T_{av,est} + 0.25\ T_{av}$

$SAIR = (w_{a1} - w_{s,m}) / (T_{a1} - T_{s,m})$

$w_{av} = SAIR \cdot (T_{av} - T_{s,m}) + w_{s,m}$

goto 7

YES : if i = NELEM
goto 16
end if
goto 5

16 At this point all elements have been analyzed. If degree of superheat or outlet quality was specified, the difference between the calculated outlet condition and the specified is compared to the acceptable tolerance. In case of specified outlet quality the convergence criteria is based on the refrigerant enthalpy at the specified quality. Since refrigerant pressure drop is accounted for in the finite element model, the change in the outlet refrigerant enthalpy, corresponding to the specified outlet quality, with pressure drop is accounted for.

if degree of superheat is specified:
if $|\Delta T_{sup} - \Delta T_{sup,spc}| < TOLER_{sup}$
goto 17
end if

if outlet quality is specified:

$i_{r,out} = i_{r,f} + x_{out} \cdot i_{r,fg}$

$i_{r,spc} = i_{r,f} + x_{spc} \cdot i_{r,fg}$

if $|i_{r,out} - i_{r,out,spc}| < TOLER_{qll}$

calculate $i_{r,f}$ and $i_{r,fg}$ corresponding to P_r(NNODE). The specified outlet enthalpy is therefore, $i_{r,out,spc} = i_{r,f} + x_{spc} \cdot i_{r,fg}$. If change in specified outlet enthalpy whithin acceptable tolerance goto 17.

end if

Estimate a new refrigerant mass flow rate, and:

goto 5.

17 The dehumidification of the air stream in each element is determined:

Do j = 1 to NELEM

18 The rate of dehumidification in element j is given by:

$$\dot{m}_w(j) = \dot{m}_a(j) \cdot (w_{a1}(i) - w_{a2}(i))$$

The latent, sensible, and total heat transfer in each element is calculated. If frost is forming on the coil, the density, the conductivity, and the thickness of the frost layer, in each element, are determined

if j = NELEM
  goto 19
end if
goto 17

19 Calculate mixed air conditions at the coil outlet. Write all results of interest to an output file.

20 If coil is frosted and time is less than specified time limit then, calculate pressure drops based on the amount of frost accumulation on the coil upto and including this time level. From fan characteristics the resulting air volumetric flow is found. The dry air mass flow is easily found from the inlet density. The constant mass flux at the next time level is therefore:

$$G_{max} = \dot{m}_a \,/\, \sum_j A_{min}(j)$$

where A_{min} is calculated taking the flow obstruction due to frost build up in element j into account. It is assumed that the time steps are small enough so that mass flows based on conditions at time t, can be used at time $t + \Delta t$.

If time < time limit , then goto 5.

21 End.

Outline of Three Region Model Program

The coil is analyzed in three different regions. The regions are determined by the state of the refrigerant:

i. Two phase region; taken from x_i to x_o, where,

$$x_o \in (0.75, 0.9)$$

ii. Transition region. A reduced refrigerant side heat transfer coefficient is assumed in this region. A smooth transition between the value of the heat transfer coefficient in the two phase region and the value in the superheated region is used. The refrigerant quality in the transition region is:

$$x \in (x_o, 1.0)$$

iii. Superheated region. The refrigerant is completely vaporized and it is assumed that no dehumidification of the air stream takes place in this region

1 The fluid inlet conditions are specified.

Air side:	T_a, φ, m_a or ΔP_{int} or V_{fc}
Refrigerant side:	h_{ri} or x_i, P_{sat} or T_{sat}, ΔT_{sup} or x_{out} or m_r

2 Coil geometry is specified.
The area calculations are carried out:

$$A_o,\ A_t,\ A_f,\ A_i,\ A_{fc},A_{min}$$

3 Initially the coil is assumed dry, and the parameter C, slope of the process line, is set to 0.

4 The fraction of the coil in region 1 is assumed.

$$AFC_1 = 0.5$$

5 Variables are initialized.

$$G_{max} = m_a / A_{min}$$

$$x_t = x_o$$

$$JFLAG = 0$$

$$KFLAG = 0$$

$$T_{a2} = T_{ai} - 2.0$$
$$T_{av} = (T_{a1} + T_{a2})/2.0$$
$$A_{min,j} = A_{min} \cdot AFC$$
$$m_{a,j} = G_{max} \cdot A_{min,j}$$

6 Calculate amount of heat transfer available in region under consideration.

$$q_{avl} = m_r \cdot i_{r,fg} \cdot (x_o - x_i)$$

7 Calculate transport properties of the air stream based upon T_{av}.

Set, $T_{av,est} = T_{av}$.

8 Calculate slope of process line, C. Coil is initially assumed dry. In other words, if first iteration set C = 0. Otherwise:

$$C = \frac{w_{av} - w_{s,m}}{T_{av} - T_{s,m}}$$

If C < 0, set C = 0.

9 From the inlet air condition and the value of C, the air side heat transfer coefficient and the fin effectiveness can be determined in the saturated refrigerant region considered.

$$h_a,\ \phi_o$$

10 Using m_r, Δx, and length of region under consideration the heat transfer coefficient on the refrigerant side is determined:

$$h_r$$

11 The thermal resistance terms can now be calculated:

refrigerant side,

$$Res_r = \frac{1}{h_r \cdot A_i \cdot AFC_j \cdot n}$$

air side,

$$Res_a = \frac{f}{h_a \cdot \left[1 + \frac{i_w \cdot C}{Le \cdot C_{p,da}}\right] \cdot A_o \cdot \phi_o \cdot n \cdot AFC_j}$$

frost layer,

$$Res_{fst} = \frac{1}{k_{fst} \cdot A_o \cdot n \cdot AFC_j}$$

Fouling (air side and refrigerant side) and contact resistances are accounted for:

$$Res_{f,a}, Res_{f,r}, Res_c$$

$$SUMR = \Sigma\ Res_i$$

thus,

$$q = \frac{(T_{av} - T_{evap})}{SUMR}$$

$$= \frac{\Delta T_{lm}}{SUMR}$$

12 If KFLAG = 1, the evaporator leaving condition is in the region under consideration. In this case length of coil in region is known (as opposed to amount of heat transfer available). The following steps are then taken :

i) estimate leaving quality (x_2)

ii) calculate h_r , and Res_r.

iii) determine, $q = \Delta T_{lm}$ / SUMR

iv) Calculate x_2. Does it compare to previous value ?

If yes, then go onto calculate surface temperatures; goto 14.

If no, go back to ii .

13 Compare q to q_{avl} . If they compare favourably goto 14 and calculate surface temperatures. If they are not equal within acceptable tolerance assume a new value for the fraction of the coil in the saturated region under consideration and go back to point 10.

14 The surface temperatures are calculated. If the surface temperature on the coil in the region under consideration is below the dew point of the air stream the air is dehumidified. The mean surface temperature is given by:

$$T_{s,m} = T_{av} - q \cdot Res_{a,s,m}$$

where, $$Res_{a,s,m} = \frac{1}{h_a \cdot A_o \cdot n \cdot AFC_j}$$

where $T_{s,m}$ is the mean surface temperature on the coil in the region under consideration.

15 If JFLAG = 0, the dew point of the air stream is compared to $T_{s,m}$.

```
If T_s,m < T_d.pt   (air is dehumidified)
  JFLAG = 1         (condensate on coil surface)
  If  T_s < 32
    JFLAG = 2       (frosted coil surface)
  end if
  goto 8
end if
if JFLAG = 0, and T_s > T_d.pt  then goto 16

If JFLAG = 0
  Is  T_s,m = T_s,m,est  ?

  NO: set  T_s,m,est = T_s,m ; goto 8.

  YES: continue to 16.
```

16 Calculate enthalpy of air stream leaving coil in region under consideration:

$$i_{a2} = i_{a1} - q / m_{a,j}$$

The leaving air stream enthalpy, i_{a2} together with the assumed process line (straight with slope C) between inlet air and coil surface condition is used to determine the state of the leaving air stream .

17 T_{a1} , T_{a2} ; and T_{sat} are used to determine the log mean temperature difference (ΔT_{lm}). The average (log mean) temperature of air as it traverses the coil is therefore:

$$T_{av} = T_{sat} + \Delta T_{lm}$$

where,

$$\Delta T_{lm} = \frac{T_{a1} - T_{a2}}{\ln\left[\dfrac{T_{a1} - T_{sat}}{T_{a2} - T_{sat}}\right]}$$

The analysis of a region is completed when convergence on the log mean average temperature in that region is reached.

Is $|T_{av} - T_{av,est}| <$ TOLERANCE

NO : set $T_{av} = 0.75\ T_{av,est} + 0.25\ T_{av}$

$SAIR = (W_{a1} - W_{s,m}) \ / \ (T_{a1} - T_{s,m})$

$W_{av} = SAIR\ (T_{av} - T_{s,m}) + W_{s,m}$

goto 7

YES : continue.

18 In case of dehumidification of air stream the condensation rate in region under consideration is calculated:

$$m_{v,j} = m_{a,j} \cdot (W_{a1} - W_{a2,j})$$

If frost is forming on the coil, the density, the conductivity, and the thickness of the frost layer, in the region under consideration, are determined

19 For the frosted coil and time > 0 the geometric location of each refrigerant region will be slightly different from what it was at the previous time level. The mass of frost in a particular region will therefore have to be adjusted in accordance with the shift in location of a particular region (frost accumulation clearly depends on location and does not move with a particular refigerant region). This is accounted for by the following scheme :

If $x_2 = x_0$, and AFC $\leq$ 1.0

DIST = AFC * LENGTH

SDIFF = DIST - D1, where D1 is DIST from previous time step.

if SDIFF > 0

$$m_{fst,1} = m_{fst,1} + \frac{\partial\, m_{fst,2}}{\partial\, L} * \text{SDIFF}$$

$$m_{fst,2} = m_{fst,2} - \frac{\partial\, m_{fst,2}}{\partial\, L} * \text{SDIFF}$$

If SDIFF < 0

$$m_{fst,2} = m_{fst,2} - \frac{\partial\, m_{fst,1}}{\partial\, L} * \text{SDIFF}$$

$$m_{fst,1} = m_{fst,1} + \frac{\partial\, m_{fst,1}}{\partial\, L} * \text{SDIFF}$$

20 Is $| x_2 - x_0 | < 10^{-3}$?

YES: implies that first region just analyzed

If AFC > 1.0

$m_{fst,1} = m_{fst,1,hold}$

KFLAG = 1.0

goto 21

end if

Store all values calculated in region 1.

Reset variables to prepare for calculalations in region 2 (the transition region) as shown:

$x_2 = 1.0$

$i_{r1} = i_{r2}$

$x_1 = x_o$

$q_{avl} = m_r \cdot i_{r,fg} \cdot n \cdot (x_2 - x_1)$

estimate the fraction of the coil in the transition region,

$AFC_2 = 1 - AFC_1$

Goto 8

NO: implies that second region just analyzed

if $AFC = AFC_1 + AFC_2 > 1.0$

KFLAG = 1.0

goto 21

end if

Calculate average conditions in the saturated region using the values stored from region 1, and those just calculated in the analysis of region 2.

Goto 22.

21 The outlet quality is equal to 1 (i.e. saturated vapor), however the area required for the evaporation of the refrigerant is greater than the area of the coil. The outlet condition must be saturated, with quality less than one. Specify length of region and calculate evaporator leaving condition.

if degree of superheat specified

$\Delta T_{sup} = 10 \cdot (1 - AFC)$

skip superheat analysis, and goto 23.

end if

$m_{fst,2} = m_{fst,2,hold}$

$AFC = 1.0 - AFC_j$

($AFC_j = 0$, if only region 1 is analyzed at this point.

$= AFC_1$ if both region 1 and 2 are analyzed at this point.)

goto 6.

22 The superheated region is analyzed. The e-NTU method for crossflow both fluids mixed is used. The following applies in the superheated region:

- $C = 0$, i.e. the superheated region is assumed dry.
- The flow areas with clean surface (no frost) are determined.
- the fraction of the coil in the superheated region is determined from,

  $AFC_{sup} = 1.0 - AFC_1 - AFC_2$
- The heat capacities and heat transfer coefficients are determined. The heat transfer effectiveness is then obtained from NTU, and the heat capacity ratio, r. The variables of interest are then determined according to:

$$q_{sup} = e \cdot C_{min} \cdot (T_{ai} - T_{sat})$$

$$T_{a2,sup} = T_{ai} - e \cdot r \cdot (T_{ai} - T_{sat})$$

$$\Delta T_{sup} = e \cdot r \cdot (T_{ai} - T_{sat})$$

23. If degree of superheat or outlet qulity is specified, the difference between the calculated outlet condition and the spedified is compared to the acceptable tolerance.

```
if degree of superheat is specified:
if |ΔT_sup - ΔT_sup,spc| < TOLER_sup
   goto 24
end if
if outlet quality is specified:
  i_r,out = i_r,f + x_out i_r,fg
  i_r,spc = i_r,f + x_spc i_r,fg
  if |i_r,out - i_r,out,spc| < TOLER_qlt
    goto 24
end if
Estimate a new refrigerant mass flow rate, and:
goto 4.
```

24 If coil is frosted:

$$time = time + \Delta t$$

Calculate mixed air conditions at the coil outlet. Calculate the coil characteristics and coil effectiveness numbers in the regions of interest. Write all results of interest to an output file.

25 If coil is frosted and time is less than specified time limit (if time is equal to time limit goto 26.) then, calculate pressure drops based on the amount of frost accumulation upto and including this time level. From fan characteristics the resulting air volumetric flow is found. The dry air mass flow is easily found from the inlet density. The constant mass flux at the next time level is therefore:

$$G_{max} = m_a / A_{min}$$

where A_{min} is calculated taking the flow obstruction due to frost build up in each region into account. It is assumed that the time steps are small enough that mass flows based on conditions at time t, can be used at time t + Δt.
Goto 5.

26 End.

APPENDIX 3

Computer Program Listings

```
C***************************************************************************.
C
C     NUMERICAL MODELING OF DIRECT EXPANSION COILS.
C
C***************************************************************************.
C
C     THIS PROGRAM IMPLEMENTS THE FINITE ELEMENT MODEL. EVAPORATORS
C     ARE ANALYZED UNDER DRY, WET, AND FROSTED COIL SURFACE
C     CONDITIONS. THE REQUIRED INPUT AND CALCULATED OUTPUT IS
C     DISCUSSED IN THE REPORT.
C
C     VARIABLE DEFINITIONS
C------------------------------------------------------------------
C
C      ARRAYS:
C AMFL(50)......MINIMUM FREE FLOW THROUGH AREA IN EACH ELEMENT
C CONDEL(50)....DEHUMIDIFICATION RATE IN EACH ELEMENT
C DELICE(50)....FROST LAYER THICKNESS IN EACH ELEMENT
C EFF(50).......FIN EFFICIENCY IN EACH ELEMENT
C HR(51)........REFRIGERANT ENTHALPY AT EACH GRID POINT.
C HTAIR(50).....AIR SIDE HEAT TRANSFER COEFFICIENT IN EACH ELEMENT
C HTREF(50).....REFRIGERANT SIDE HEAT TRANSFER COEFFICIENT IN EACH
C               ELEMENT
C MFAET(50).....AIR MASS FLOW IN EACH ELEMENT
C PR(51)........REFRIGERANT PRESSURE AT EACH GRID POINT
C Q(50).........HEAT TRANSFERRED FROM AIR TO REFRIGERANT IN EACH
C               ELEMENT
C QLAT(50)......LATENT HEAT TRANSFER IN EACH ELEMENT
C RESICE(50)....FROST LAYER THERMAL RESITANCE IN EACH ELEMENT
C RESIST(50,3)..THERMAL RESISTANCES IN EACH ELEMENT
C SS(51)........GEOMETRIC POSITION OF GRID POINTS
C TFR(50).......MEAN SURFACE TEMPERATURE IN EACH ELEMENT
C TMCOND(50)....TOTAL DEHUMIDIFICATION IN EACH ELEMENT
C TR(51)........REFRIGERANT TEMPERATURE AT EACH GRID POINT.
C TW(50)........MEAN COIL COOLING TEMP. IN EACH ELEMENT
C V(51).........REFRIGERANT SPECIFIC VOLUME AT EACH GRID POINT.
C VLOC(50)......LOCAL AIR VALOCITY IN EACH ELEMENT
C VREF(51)......REFRIGERANT VELOCITY AT EACH GRID POINT
C X(51).........REFRIGERANT QUALITY AT EACH GRID POINT.
C Z(9)..........AIR PROPERTIES FOR USE IN SUBR. PSYC
C
```

```
C TA1(50,I).....INCOMING CROSS FLOW AIR PROPERTIES IN EACH ELEMENT
C TA2(50,I).....LEAVING CROSS FLOW AIR PROPERTIES  IN EACH ELEMENT
C
C    I=  1        2        3        4        5        6        7        8
C        DB       WB       RH       W        H        D.PT.    LH       SH
C
C
C
C ACFROST..ACCUMULATED FROST
C AFIN.....FIN SURFACE AREA
C AFINEL...FIN AREA PER ELEMENT
C AFC......FACE AREA OF COIL
C AIEL.....TUBE INSIDE SURFACE AREA PER ELEMENT.
C AMINF....MINIMUM AIR FLOW THROUGH AREA
C AMINFL...MINIMUM AIR FREE FLOW THROUGH AREA IN AN ELEMENT
C ATB......TUBE SURFACE AREA EXPOSED TO AIR
C ATBEL....TUBE SURFACE AREA PER ELEMENT
C ATOT.....TOTAL AIR SIDE HEAT TRANSFER AREA.
C ATOTEL...TOTAL AIR SIDE SURFACE AREA IN AN ELEMENT
C AX.......TUBE INSIDE CROSS-SECTIONAL AREA.
C CKTS.....NUMBER OF CIRCUITS (OR ROWS)
C CONDFR...THERMAL CONDUCTIVITY OF FROST LAYER
C CPA......HEAT CAPACITY OF THE AIR.
C CPAD.....HEAT CAPACITY OF DRY AIR
C CPR......HEAT CAPACITY OF THE REFRIGERANT
C CPRLS....HEAT CAPACITY OF THE REFRIGERANT IN LIQUIDE SATURATED STATE
C DC.......DEPTH OF ONE ROW (LONGITUDINAL SPACING)
C DENICE...DENSITY OF ICE
C DENSDA...DENSITY OF DRY AIR
C DENSFR...FROST DENSITY
C DEQUIV...EQUIVALENT FIN DIAMETER
C DH ......HYDRAULIC DIAMETER ON THE AIR SIDE.
C DHEL.....HYDRAULIC DIAMETER IN AN ELEMENT
C DI.......INSIDE TUBE DIAMETER.
C DO.......OUTSIDE TUBE DIAMETER.
C DPINIT...INITIAL NONFROSTED COIL PRESSURE DROP
C DPTH.....COIL DEPTH
C DTIME....TIME STEP
C DTLM.....LOG MEAN TEMPERATURE DIFFERENCE
C DTSHSP...SPECIFIED DEGREE OF SUPERHEAT AT EVAPORATOR OUTLET
C FINS.....NUMBER OF FINS.
C FH.......FIN HEIGHT
C FHEQIV...EQUIVALENT FIN HEIGHT
```

```
C FPI......FIN DENSITY (FIN PER INCH)
C HA2......AIR LEAVING ENTHALPY
C HC.......HEIGHT OF ONE ROW (LATTITUDINAL ROW SPACING)
C HCONT....CONTACT CONDUCTANCE
C HFGW.....HEAT OF CONDENSATION OR DESUBLIMATION
C HFOULI...INSIDE FOULING CONDUCTANCE (REF. SIDE)
C HFOULO...OUTSIDE FOULING CONDUCTANCE (AIR SIDE)
C HT.......HEIGHT OF COIL
C HTA......AIR SIDE HEAT TRANSFER COEFFICIENT
C HTAD.....SENSIBLE HEAT TRANSFER COEFFICENT
C HTAW.....LATENT HEAT TRANSFER COEFFICIENT
C HTR......REFRIGERANT SIDE HEAT TRANSFER COEFFICIENT
C HWALL....SATURATED AIR HUMIDITY AT SURFACE TEMPERATURE
C I........COUNTS ELEMENTS
C INDIC....FLAG. IF 1, SUPERHEAT SPECIFIED. IF 2, QUALITY SPECIFIED
C INTEV....FLAG. IF 1, EVAPORATOR TEMPERATURE SPECIFIED
C INDQL....FLAG. IF 1, INLET QUALITY IS SPECIFIED
C INDVEL...FLAG. IF 1, THE LOCAL VELOCITY IS SPECIFIED
C INDVFC...FLAG. IF 1, THE AIR FACE VELOCITY IS SPECIFIED
C JFLAG....INDICATES COIL SURFACE CONDITION
C          =1, CONDENSATE ON COIL SURFACE
C          =2, FROST ON COIL SURFACE
C KA.......THERMAL CONDUCTIVITY OF AIR
C KFIN.....THERMAL CONDUCTIVITY OF FIN MATERIAL.
C KR.......THERMAL CONDUCTIVITY OF REFRIGERANT.
C KRLS.....THERMAL CONDUCTIVITY OF REFRIGERANT IN LIQUID SATURATED
C          STATE.
C LE.......LEWIS NUMBER
C LENGTH...LENGTH OF COIL.
C MF2PA....AIR MASS VELOCITY
C MF2PR....REFRIGERANT MASS VELOCITY
C MFA......MASS FLOW OF AIR.
C MFAEL....MASS FLOW OF AIR IN A PARTICULAR ELEMENT
C MFROST...TOTAL MASS OF FROST
C MFR......MASS FLOW OF REFRIGERANT PER CIRCUIT
C MFRT.....TOTAL REFRIGERANT MASS FLOW RATE
C MUA......DYNAMIC VISCOSITY OF AIR.
C MUR......DYNAMIC VISCOSITY OF REFRIGERANT.
C MURLS....DYNAMIC VISCOSITY OF REFRIGERANT IN LIQUID SATURATED STATE
C MURVS....DYNAMIC VISCOSITY OF REFRIGERANT IN VAPOR SATURATED STATE
C NELEM....NUMBER OF ELEMENTS IN THE GRID.
C NF.......AIR SIDE FIN EFFICIENCY.
C NNODE....NUMBER OF NODES (GRID POINTS)
```

```
C NREF.....REFRIGERANT NUMBER
C PI.......PI, 3.14159...
C PINTVL...PRINTING INTERVAL
C PRA......AIR PRANDTL NUMBER.
C QIN......INLET QUALITY
C QLTYSP...SPECIFIED EVAPORATOR OUTLET QUALITY
C QTOT.....TOTAL HEAT TRANSFERRED IN THE HEAT EXCHANGER.
C RESA.....THERMAL RESISTANCE ON THE AIR SIDE
C RESCON...CONTACT THERMAL RESISTANCE
C RESFLI...INSIDE FOULING RESISTANCE
C REFLO....OUTSIDE FOULING RESISTANCE
C RESR.....THERMAL RESISTANCE OF THE REFRIGERANT SIDE
C RH1......RELATIVE HUMIDITY OF INLET AIR STREAM
C RORLS....DENSITY OF SATURATED LIQUID REFRIGERANT.
C RORVS....DENSITY OF SATURATED VAPOR REFRIGERANT.
C ROWS.....NUMBER OF ROWS (EQUIVALENT TO CIRCUITS)
C RWSH.....ROWS HIGH (PASSES PER CIRCUIT)
C S........INTERSPATIAL FIN DISTANCE
C SUMR.....SUMMATION OF THERMAL RESISTANCES
C TAV......AVERAGE (LOG MEAN) AIR TEMPERATURE
C TEVAP....EVAPORATOR TEMPERATURE
C TLIMIT...TIME LIMIT
C VFACE....FACE VELOCITY
C VSPEC....SPECIFIED LOCAL VELOCITY
C WAV......AVERAGE SPECIFIC HUMIDITY (CORRESPONDING TO TAV)
C WDTH.....WDTH OF COIL (LENGTH PER CIRCUIT PASS)
C XO.......INLET QUALITY TO THE TRANSITION REGION
C X2APX....ESTIMATE OF QUALITY OF REFRIGERANT AT 2.
C Y........FIN THICKNESS.
C
C*********************************************************************
C
      PROGRAM EVAP(TAPE10,TAPE20,TAPE30)
      INTEGER I,JCR,K,N,NR
      REAL SS(51),TR(51),PR(51),X(51),V(51),HR(51),Z(9)
      REAL HTAIR(50),HTREF(50),EFF(50),TW(50),CONDEL(50),Q(50)
      REAL TA1(50,9),TA2(50,9),AA(3,8),RESIST(50,3),DELICE(50)
      REAL RESICE(50),VLOC(50),AMFL(50),MFAET(50),QLAT(50),TWS(50)
      REAL TMCOND(50),VREF(51),TFR(50),QRATS(50),DPQRS(50),REYNS(50)
      REAL KFIN,KR,KRLS,LENGTH,MALP,MF2PA,MF2PR,MFA,MFAEL,MFR
      REAL MUA,MUR,MURLS,MURVS,NF,KA,MRHS,MLHS,MFROST,LE,MFRT
```

```
C-----------------------------------------------------------------
C  THE PROPERTIES OF INCOMING AIR AND REFRIGERANT ARE SPECIFIED.
C-----------------------------------------------------------------
      READ(10,*) NREF
      READ(10,*) NNODE,NELEM,XO
      READ(10,*) TA1(1,1),RH1,MFA,DPINIT,CPAD,HFGW
      READ(10,*) TR(1),HR(1),PR(1),MFRT
      READ(10,*) DTIME,TLIMIT,PINTVL
      READ(10,*) DTSHSP,QLTYSP,INDIC
      READ(10,*) INDQL,QIN
      READ(10,*) INDTEV,TEVAP
      READ(10,*) INDVEL,VSPEC
      READ(10,*) INDVFC,VFACE
C-----------------------------
C  FIN GEOMETRY IS SPECIFIED
C-----------------------------
      READ(10,*) AFC,WDTH,HT,DPTH,CKTS,RWSH
      READ(10,*) DO,DI,Y,FH,KFIN,FPI
      READ(10,*) HCONT,HFOULI,HFOULO
      ROWS=CKTS
      HC=HT/RWSH
      DC=DPTH/ROWS
      LENGTH=RWSH*WDTH
      FINS=FPI*LENGTH*12
      S=1.0/(FPI*12)
C-----------------------------------------------------------------------
C VARIABLES USED IN THE PROPERTY SUBROUTINES ARE INITIALIZED BY
C CALLING THE FOLLOWING DATA SUBROUTINES.
C-----------------------------------------------------------------------
      CALL DATA(NREF)
      CALL DATAV(NREF)
      CALL DATAK(NREF)
      CALL DATASH(NREF)
C
      IF (INDTEV .EQ. 1) THEN
       TR(1)=TEVAP
      ELSE
       TR(1)=TSAT(ITR,PR(1))
      END IF
      CALL SATPRP(ITR,TR(1),PR(1),VF,VG,HF,HFG,HG,SF,SG)
      IF (INDQL .EQ. 1) HR(1)=HF+QIN*HFG
      QIN=(HR1-HF)/HFG
      Z(1)=TA1(1,1)
```

```
      Z(3)=RH1
C-----------------------------------------------------------------------
C  AIR PROPERTIES ARE EVALUATED. GIVEN DRY BULB TEMPERATURE AND
C  RELATIVE HUMIDITY, THE REMAINING AIR PROPERTIES OF INTEREST
C  CAN BE DETERMINED.
C-----------------------------------------------------------------------
      CALL PSYC(Z,3)
      DO 20 K=1,9
         DO 15 J=1,50
            TA1(J,K)=Z(K)
   15    CONTINUE
   20 CONTINUE
      DENSA1=1.0/Z(9)
      KA=-8.487E-08*Z(1)*Z(1)+3.1923E-05*Z(1)+1.3005E-02
      PRA=1.047E-06*Z(1)*Z(1)-2.3187E-04*Z(1)+0.72307
      MUA=-1.962E-07*Z(1)*Z(1)+8.2371E-05*Z(1)+3.9207E-02
      DENSDA=DENSA1
C-------------------------------------
C  VARIABLES ARE INITIALIZED
C-------------------------------------
      LE=0.95
      FINSPC=S-Y
      AFACE=LENGTH*HC
      DP=DPINIT
      CFM=-534.03*DP+1560.7
      TMFA=DENSA1*CFM*60
      IF (TMFA .LT. MFA) MFA=TMFA
      IF (INDVFC .EQ. 1) MFA=60*DENSA1*VFACE*AFACE
      VFACE=MFA/(60*DENSA1*AFACE)
      WSP=VFACE*S/(S-Y)
      MFR=MFRT/ROWS
      JJ2=0
      DTSH=0
      J1=0
      K1=0
      JFLAG=0
      QSUM=0
      KK=0
      QTOT=0
      TCOUNT=0
      ACFROST=0
      DENICE=0.9*62.4
      DO 25 JJ=1,50
```

```
      RESICE(JJ)=0
   25 DELICE(JJ)=0
      JCR=0
      SS(1)=0
      TIME=0
      PI=2.0*ATAN2(1.0,0.0)
      XLONG=LENGTH/NELEM
      DO 30 J=2,NNODE
   30 SS(J)=SS(J-1)+XLONG
      X(1)=(HR(1)-HF)/HFG
      DQDEL=MFR*(HG-HR(1))/NELEM
      DXDL=(1.0-X(1))/NELEM
      X(2)=X(1)+DXDL*XLONG
      HREFSP=HF+QLTYSP*HFG
      COEFF=0.05
      CALL FLAREA(DO,DELICE(1),PI,Y,FINS,HC,DC,LENGTH,
     & AFRONT,ATL,ATBEL,AFINEL,AMINF,DH,NELEM,DEQUIV,FHEQIV)
C-----------------------------------------------------------
C  AN INITIAL WALL TEMPERATURE IS ASSUMED
C-----------------------------------------------------------
      TA2(1,1)=TA1(1,1)-2.0
      TA2(1,4)=TA1(1,4)
      TW(1)=(TA1(1,1)+TR(1))/2
      TFR(1)=TW(1)
   35 CONTINUE
      DXDL=(X(2)-X(1))/XLONG
      TAV=(TA1(1,1)+TA2(1,1))/2
      WAV=(TA1(1,4)+TA2(1,4))/2
C-----------------------------------------------------------------------------
C  INITIALLY NO CONDENSATION IS ASSUMED AND THE SLOPE OF THE
C  PSYCROMETRIC PROCESS LINE, C, IS THEREFORE SET TO 0.
C-----------------------------------------------------------------------------
      C=0
C-----------------------------------------------------------------------------
C  AREAS AND FLOW RATES USED IN THE SUBROUTINES EVALUATING HEAT
C  TRANSFER COEFFICIENTS ARE DETERMINED.
C-----------------------------------------------------------------------------
      AFRONT=HC*LENGTH
      AIEL=PI*DI*LENGTH/NELEM
      AOEL=PI*DO*LENGTH/NELEM
      AX=PI*(DI*DI)/4.0
      MF2PR=MFR/AX
      MFAEL=MFA/NELEM
```

```
      AMFTOT=(HC-DO)*(LENGTH-Y*FINS)
      IF (INDVEL .EQ. 1) MFA=DENSA1*AMFTOT*VSPEC*60
      MF2PA=MFA/AMFTOT
      RESCON=1.0/(HCONT*AOEL*ROWS)
      RESFLI=1.0/(HFOULI*AIEL*ROWS)
      RESFLO=1.0/(HFOULO*ATL*ROWS)
C
   40 CALL FLAREA(DO,DELICE(1),PI,Y,FINS,HC,DC,LENGTH,
     & AFRONT,ATOTEL,ATBEL,AFINEL,AMINF,DH,NELEM,DEQUIV,FHEQIV)
      VFRONT=MFA/(DENSDA*AFRONT*3600)
      MFAEL=MF2PA*AMINF
      SIGMA=AMFTOT/AFRONT
      ASAT=4*DC*HC*SIGMA/(DO*DH*PI)
      VFC=MFAEL/(DENSDA*AMINF*3600)
      DO 45 JJ=1,NELEM
      MFAET(JJ)=MFAEL
   45 VLOC(JJ)=VFC*60
      JFLAG=0
      QSUM=0
C ********************************************************************
C AN ITERATIVE PROCEDURE TO OBTAIN THE AIR AND REFRIGERANT STATES AT
C PRESCRIBED NODES AND ELEMENTS ALONG THE HEAT EXCHANGER ROW IS USED
C ********************************************************************
      DO 85 I=1,NELEM
      KK2=0
      KKOUNT=0
      KK=0
      IF (JFLAG .EQ. 2) THEN
      CALL FLAREA(DO,DELICE(I),PI,Y,FINS,HC,DC,LENGTH,
     & AFRONT,ATOTEL,ATBEL,AFINEL,AMINF,DH,NELEM,DEQUIV,FHEQIV)
      IF (AMINF .LT. 0) AMINF=0
      SIGMA=AMINF*NELEM/AFRONT
      ASAT=4*DC*HC*SIGMA/(DO*DH*PI)
      MFAEL=MF2PA*AMINF
      MFAET(I)=MFAEL
      VLOC(I)=MFAEL/(DENSDA*AMINF*60)
      END IF
C
C THE WALL TEMP IS INITIALLY ASSUMED TO STAY CONSTANT FROM ELEMENT
C I TO I+1.
C---------------------------------------------------------------------
      IF (I .NE. 1) TW(I)=TW(I-1)
C
```

```
C THE CONVERGENCE CRITERIA IN EACH ELEMENT IS ON THE AVERAGE
C TEMPERATURE.
C------------------------------------------------------------------
   50 TAVEST=TAV
C
C THE TRANSPORT PROPERTIES OF AIR AT THE AVERAGE TEMPERATURE IS
C EVALUATED.
C------------------------------------------------------------------
      KA=-8.487E-08*TAV*TAV+3.1923E-05*TAV+1.3005E-02
      PRA=1.047E-06*TAV*TAV-2.3187E-04*TAV+0.72307
      MUA=-1.962E-07*TAV*TAV+8.2371E-05*TAV+3.9207E-02
      CPA=CPAD+0.45*WAV
      Z(1)=TAV
      Z(4)=WAV
      CALL PSYC(Z,4)
      DENSDA=1.0/Z(9)
      VLOC(I)=MFAEL/(DENSDA*AMINF*60)
C
   55 TWEST=TW(I)
      IF (JFLAG .EQ. 0) GOTO 60
      Z(1)=TFR(I)
      Z(3)=1.0
      CALL PSYC(Z,3)
      WFR=Z(4)
C------------------------------------------------------------------
C THE SLOPE OF THE ASSUMED PSYCROMETRIC PROCESS LINE IS CALCULATED
C AND THE AIR AND REFRIGERANT SIDE HEAT TRANSFER COEFFICIENTS ARE
C DETERMINED.
C------------------------------------------------------------------
      C=(WAV-Z(4))/(TAV-TFR(I))
      IF (C .LT. 0) C=0
   60 CALL HEAT2(CPA,PRA,MF2PA,MUA,DO,ASAT,HTAD,DC)
      HTAW=HTAD*HFGW*C/(CPA*LE)
      HTA=HTAD+HTAW
      CALL FINEF3(HTA,KFIN,HFGW,Y,DO,DEQUIV,FHEQIV,NF)
      RESA=1/(HTA*(ATBEL+NF*AFINEL)*ROWS)
      RESFM=1.0/(HTA*ATOTEL*ROWS)
      LL=0
   65 CALL CALC(JCR,I,MF2PR,DI,AIEL,TEVAL,DXDL,DQDEL,X,V,HR,
     & PR,TR,HTR,RESR,NNODE,SS,ROWS,PI,XO)
C
      SUMR=RESR+RESA+RESICE(I)+RESCON+RESFLI+RESFLO
      Q(I)=(TAV-TR(I))/SUMR
```

```
C-----------------------------------------------------------------------
C A NEW ESTIMATE OF THE CHANGE IN REF. QUALITY IN ONE ELEMENT IS
C MADE, AND COMPARED TO THE PREVIOUS ESTIMATE, DXDL.
C THE STATE OF THE REFRIGERANT LEAVING THE ELEMENT IS CALCULATED.
C-----------------------------------------------------------------------
      HR(I+1)=HR(I)+Q(I)/(MFR*ROWS)
      CALL REFLV(JCR,I,MF2PR,X,V,HR,PR,TR,NNODE,DI,XLONG)
      DIFF=X(I+1)-X(I)
      IF (ABS(DIFF-DXDL) .GT. 0.000002) THEN
         LL=LL+1
         DXDL=DIFF
         DQDEL=Q(I)/ROWS
         IF (LL .LT. 15) GOTO 65
      END IF
      IF (X(I+1) .GE. 1.0) X(I+1)=1.000001
      TW(I)=TAV-Q(I)*(RESFM+RESICE(I))
      TFR(I)=TAV-Q(I)*RESFM
C-----------------------------------------------------------------
C THE CALCULATED WALL TEMP IS COMPARED TO THAT CALCULATED IN
C PREVIOUS ITERATION.
C-----------------------------------------------------------------
      KK=KK+1
      IF (KK .GT. 30) GOTO 70
      IF (ABS(TWEST-TW(I)) .GT. 0.005) GOTO 55
   70 IF (JFLAG .EQ. 0) THEN
      IF (TW(I) .LT. TA1(I,6)) THEN
        JFLAG=1
        IF ( (TW(I)-32.0) .LT. 0.001) THEN
         JFLAG=2
         HFGW=1219.0
        END IF
        GOTO 55
      END IF
      END IF
C-------------------------------------------
C LEAVING AIR ENTHALPY IS CALCULATED.
C-------------------------------------------
      HA2=TA1(I,5)-Q(I)/MFAEL
C-----------------------------------------------------------------------
C THE STATES OF THE AIR ARE ASSUMED TO FOLLOW A STRAIGHT LINE ,OF
C SLOPE C, ON THE PSYCROMETRIC CHART, BETWEEN INCOMING STATE AND
C SATURATED STATE OF AIR NEXT TO THE COIL SURFACE. KNOWING HA2
C AND THE PROCESS LINE, TA2 CAN BE FOUND USING A BISECTION METHOD.
```

```
C------------------------------------------------------------------
      CALL TLEAV2(I,KK2,HA2,C,TW,TA1,Z,HWALL,NELEM)
C------------------------------------------------------------------
C CONVERGENCE ON THE AVERAGE AIR TEMPERATURE IS MONITORED.
C------------------------------------------------------------------
      DTLM=(TA1(I,1)-Z(1))/ALOG((TA1(I,1)-TR(I))/(Z(1)-TR(I)))
      TAV=TR(I)+DTLM
      SAIR=(TA1(I,4)-WFR)/(TA1(I,1)-TFR(I))
      WAV=SAIR*(TAV-TFR(I))+WFR
      KKOUNT=KKOUNT+1
      IF (KKOUNT .GT. 20) THEN
       WRITE(20,*) 'I=',I,' KKOUNT REACHES 20'
       GOTO 75
      END IF
      IF (ABS(TAV-TAVEST) .GT. 0.005) THEN
       IF (KKOUNT .EQ. 1) TAVEST=TAV
       TAV=(0.25*TAV+0.75*TAVEST)
       WAV=SAIR*(TAV-TFR(I))+WFR
       GOTO 50
      END IF
   75 IF (HWALL .GT. HA2) THEN
       WRITE(30,*) 'HA2 IS LESS THAN HWALL.'
       WRITE(30,*) 'I=',I
       WRITE(30,*) 'HA2=',HA2,' HWALL=',HWALL
      END IF
C------------------------------------------------------------------
C HEAT TRANSFER PROPERTIES ARE STORED IN ARRAY.
C------------------------------------------------------------------
      HTAIR(I)=HTA
      HTREF(I)=HTR
      EFF(I)=NF
      RESIST(I,1)=RESA
      RESIST(I,3)=RESR
      DO 80 K=1,9
         TA2(I,K)=Z(K)
   80 CONTINUE
   85 CONTINUE
C------------------------------------------------------------------
C THE PERCENTAGE OF THE COIL IN THE 2 PHASE REF.REGION IS CALCULATED.
C------------------------------------------------------------------
      DO 90 J=1,NNODE
      IF (X(J) .GE. 1.0) THEN
       PNODE=J
```

```
        GOTO 95
      END IF
   90 CONTINUE
      PNODE=NNODE
   95 PERC=(PNODE/NNODE)*100.0
C
C IF ITERATIONS ARE MADE ON THE DEGREE OF SUPERHEAT, OR OUTLET QUALITY
C A NEW REFRIGERANT FLOW RATE IS ESTIMATED.
C-----------------------------------------------------------------------
      IF (INDIC .EQ. 0) GOTO 105
      IF (INDIC .EQ. 1) THEN
        DTSH=TR(NNODE)-TR(1)
        DIFF=DTSH-DTSHSP
        IF (ABS(DIFF) .LE. 1.0) THEN
          J1=0
          K1=0
          COEFF=0.05
          GOTO 105
        END IF
      ELSE
        DTSH=HR(NNODE)
        DIFF=DTSH-HREFSP
        DTSHSP=HREFSP
        IF (ABS(DIFF) .LE. 0,5) GOTO 100
      END IF
      IF (DIFF .LT. 0) THEN
        MRHS=MFR
        DTSHR=DTSH
        J1=1
      ELSE
        MLHS=MFR
        DTSHL=DTSH
        K1=1
      END IF
      IF (JJ4 .GE. 3) THEN
       MFR=(MRHS+MLHS)/2
       GOTO 35
      END IF
C
C CONVERGENCE PROBLEMS ARE SOMETIMES ENCOUNTERED DUE TO THE FINITE
C SIZE OF THE ELEMENTS. IF CONVERGENCE IS NOT ATTAINED IN 10 STEPS
C THE PROGRAM IS TERMINATED.
C-----------------------------------------------------------------------
```

```
      JJ2=JJ2+1
      IF (JJ2 .GT. 10) GOTO 270
      IF ((J1 .EQ. 1) .AND. (K1 .EQ. 1)) THEN
        MFR=(DTSHSP-DTSHR)*(MLHS-MRHS)/(DTSHL-DTSHR)+MRHS
        JJ4=JJ4+1
        GOTO 35
      END IF
      IF (DIFF .LT. 0) THEN
        MFR=MRHS-COEFF*MRHS
      ELSE
        MFR=MLHS+COEFF*MLHS
      END IF
      GOTO 35
C
C THE ENTHALPY CORRESPONDING TO THE SPECIFIED OUTLET QUALITY IS A
C FUNCTION OF THE PRESSURE DROP. HENCE CONVERGENCE ON ENTHALPY NEED
C TO BE CHECKED.
C-----------------------------------------------------------------------
  100 P2=PR(NNODE)
      T2=TSAT(ITR,P2)
      CALL SATPRP(ITR,T2,PSAT,VF,VG,HF,HFG,HG,SF,SG)
      HNEW=HF+QLTYSP*HFG
      HREFSP=HNEW
      J1=0
      K1=0
      COEFF=0.05
      IF (ABS(HNEW-HREFSP) .LE. 0.5) GOTO 105
      GOTO 35
  105 CONTINUE
C-----------------------------------------------------------------------
C THE DEHUMIDIFICATION RATE IN ALL ELEMENTS ARE CALCULATED.
C-----------------------------------------------------------------------
      DO 110 I=1,NELEM
      CONDEL(I)=MFAET(I)*(TA1(I,4)-TA2(I,4))
      QLCOMP=MFAET(I)*(TA1(I,8)-TA2(I,8))
      QLAT(I)=CONDEL(I)*HFGW
      KK=0
      QSUM=QSUM+Q(I)
C-----------------------------------------------------------------------
C IN CASE OF FROSTING ON THE COIL THE ACCUMULATED FROST IN TIME DT,
C AND THE RESULTING RESISTANCE TO HEAT TRANSFER IS DETERMINED AT
C ELEMENT I.
C-----------------------------------------------------------------------
```

```
      IF (JFLAG .EQ. 2) THEN
      IF (CONDEL(I) .GT. 0) THEN
       TMCOND(I)=TMCOND(I)+CONDEL(I)*DTIME/ROWS
       CALL FROST2(I,JCR,NELEM,TMCOND,DELICE,RESICE,Q,QLAT,S,Y,
     & TW(I),ATL,VSPEC,DENSDA,MUA,DENICE,TIME,DTIME,ROWS,VFACE,INDVEL,
     & QRATS,DPQRS,REYNS,TWS)
      END IF
      RESIST(I,2)=RESICE(I)
      DIAO=DO+DELICE(I)
      AMFL(I)=(HC-DIAO)*(LENGTH-(Y+2*DELICE(I))*FINS)/NELEM
      IF (AMFL(I) .LT. 0) AMFL(I)=0
      END IF
C-----------------------------------------------------------------
C JCR IS AN ERROR CONTROL CHARACTER USED IN THE PROGRAM
C-----------------------------------------------------------------
      IF (JCR .GT. 0)THEN
         WRITE(20,260) JCR
         GOTO 270
      END IF
  110 CONTINUE
C
C REFRIGERANT VELOCITY IS DETERMINED.
C-----------------------------------------
      DO 115 JK=1,NNODE
       VREF(JK)=V(JK)*MFR/(AX*60.0)
  115 CONTINUE
      MFRT=MFR*ROWS
C
      IF (TIME .GT. 0.001) GOTO 120
      ICTR=1
      WRITE(20,135) ICTR
      WRITE(20,155)
      WRITE(20,160)
      WRITE(20,210)
      WRITE(20,170)
      WRITE(20,215) AFC,WDTH,HT,DPTH
      WRITE(20,220) FPI,RWSH,CKTS
      WRITE(20,225) HC,DC,DO,DI
      WRITE(20,230) Y,S,FH,FINS
      WRITE(20,235) ROWS,LENGTH,KFIN
      WRITE(20,165)
      WRITE(20,170)
      WRITE(20,175) TA1(1,1),RH1,PRA,KA
```

```
      WRITE(20,180) MFA,MUA,HFGW
      WRITE(20,185) TR(1),HR(1),PR(1),MFRT
      IF (INDIC .EQ. 0) WRITE(20,190) MFRT
      IF (INDIC .EQ. 1) WRITE(20,195) DTSHSP
      IF (INDIC .EQ. 2) WRITE(20,200) QLTYSP
      WRITE(20,205) AFACT,RFACT
C----------------------------------------------------------------------
C IN THE CASE OF DRY OR WET TUBES AVG RESULTS ARE CALCULATED AND
C ALL RESULTS ARE SENT TO AN OUTPUT FILE.
C----------------------------------------------------------------------
  120 CONTINUE
      IF (JFLAG .NE. 2) THEN
        QTOTAL=MFR*(HR(I)-HR(1))*ROWS
CC      WRITE(20,1750) QSUM
CC      WRITE(20,1740) QTOTAL
        WRITE(20,135) ICTR
        IF (JFLAG .EQ. 0) WRITE(20,140)
        IF (JFLAG .EQ. 1) WRITE(20,145)
        WRITE(20,160)
        CALL OVERAL(MFAEL,NELEM,TA1,TA2,TW,CONDEL,TIME,DELICE,
     & PR(NELEM),X,JFLAG,MFAET,MFA,ACFROST,INDIC,PERC,MFRT,TFR,HFGW)
        CALL RSLTS(TA1,TA2,Q,TW,TR,PR,X,HTAIR,HTREF,SS,HR,
     & EFF,CONDEL,RESIST,DELICE,VLOC,NNODE,NELEM,VREF,TFR)
        GOTO 270
      END IF
C----------------------------------------------------------------------
C THE TUBES ARE FROSTED. THIS TRANSIENT PROCCES IS ANALYZED ASSUMING
C A QUASI STEADY STATE AT EACH TIME LEVEL DT.
C----------------------------------------------------------------------
  125 AMFTOT=0
      CONDT=0
      DO 130 J=1,NELEM
      CONDT=CONDT+CONDEL(J)
  130 AMFTOT=AMFTOT+AMFL(J)
      MFROST=CONDT*DTIME
      ACFROST=ACFROST+MFROST
C
      QTOTAL=MFR*(HR(I)-HR(1))*ROWS
      TCOUNT=TCOUNT+DTIME
      TIME=TIME+DTIME
      IF ((TCOUNT-PINTVL) .GE. -0.0001) THEN
CC    WRITE(20,1750) QSUM
CC    WRITE(20,1740) QTOTAL
```

```
      WRITE(20,135) ICTR
      WRITE(20,150)
      WRITE(20,160)
      CALL OVERAL(MFAEL,NELEM,TA1,TA2,TW,CONDEL,TIME,DELICE,
     & PR(NELEM),X,JFLAG,MFAET,MFA,ACFROST,INDIC,PERC,MFRT,TFR,HFGW)
      CALL RSLTS(TA1,TA2,Q,TW,TR,PR,X,HTAIR,HTREF,SS,HR,
     & EFF,CONDEL,RESIST,DELICE,VLOC,NNODE,NELEM,VREF,TFR)
      TCOUNT=0
      END IF
C
C THE AIR FLOW RATE IS CORRELATED WITH THE AMOUNT OF FROST ON THE COIL
C-----------------------------------------------------------------------
      CALL PDROP(NELEM,DO,DELICE,PI,Y,FINS,HC,DC,LENGTH,AFRONT,
     & MF2PA,MUA,DENSDA,ACFROST,MFA,S,DPAIR,ROWS,VFACE,HBAR)
      DP=ROWS*DPAIR
      DP2=DP
      IF (DP2 .LT. DPINIT) DP2=DPINIT
      CFM=-534.03*DP2+1560.7
      RMFA=DENSA1*CFM*60
      IF (INDVEL .EQ. 1) RMFA=60*VSPEC*DENSA1*AMFTOT
      IF (INDVFC .EQ. 1) THEN
       CFM=-534.03*DP+1560.7
       RMFA=DENSA1*CFM*60
       SMFA=60*DENSA1*VFACE*AFACE
       IF (SMFA .LT. RMFA) RMFA=SMFA
      END IF
      IF (RMFA .LT. MFA) MFA=RMFA
      VFACE=MFA/(60*DENSA1*AFACE)
      MF2PA=MFA/AMFTOT
      DXDL=(X(2)-X(1))/XLONG
      JJ4=0
      IF (TIME .LT. TLIMIT) GOTO 40
C-----------------------------------------------------------
C FORMATTING OUTPUT
C-----------------------------------------------------------
  135 FORMAT(I1)
  140 FORMAT(30X,'THE COIL IS DRY')
  145 FORMAT(30X,'THE COIL IS WET')
  150 FORMAT(30X,'THE COIL IS FROSTED')
  155 FORMAT(25X,'I N P U T   P A R A M E T E R S')
  160 FORMAT(28X,'----------------------------',/)
  165 FORMAT(15X,'ENTERING AIR AND REFRIGERANT STATES')
  170 FORMAT(13X,'---------------------------------------------',/)
```

```
175   FORMAT(10X,'TA1=',F5.2,1X,'F',3X,'RH=',F5.3,2X,'PRA=',F5.3,1X,
     & 'BTU/LB*F',3X,'KA=',F5.2,1X,'BTU/HR*F*FT',/)
180   FORMAT(10X,'MFA=',F7.1,1X,'LB/HR',3X,'MUA=',F6.4,1X,'LB/FT*HR',
     & 3X,'HFGW=',F6.1,1X,'BTU/LB',/)
185   FORMAT(10X,'TR=',F5.2,1X,'F',3X,'HR(1)=',F6.2,3X,'PR(1)=',F5.2,
     & 1X,'PSI',3X,'MFR=',F7.2,1X,'LB/HR',/)
190   FORMAT(10X,'SPECIFIED REFR. MASS FLOW=',F7.2,' LB/HR',/)
195   FORMAT(10X,'SPECIFIED DEGREE OF SUP.HEAT=',F6.2,' F',/)
200   FORMAT(10X,'SPECIFIED LVG.REF.QUALITY=',E10.3,/)
205   FORMAT(10X,'AIR RES.FACT.=',F4.2,4X,'REF.RES.FACT.=',F4.2,//)
210   FORMAT(15X,'GEOMETRIC DATA. ALL DATA IN UNITS OF FEET.')
215   FORMAT(10X,'FACE A.=',F6.3,2X,'WIDTH=',F6.3,2X,'HEIGHT=',
     & F6.3,2X,'DEPTH=',F6.3,/)
220   FORMAT(10X,'FPI=',F4.1,2X,'ROWS H.=',F4.1,2X,'CKTS=',F4.1,/)
225   FORMAT(10X,'ROW HEIGHT=',F6.3,2X,'ROW DEPTH=',F6.3,
     & 2X,'DIA.O.=',F6.4,2X,'DIA.I.=',F6.4,/)
230   FORMAT(10X,'FIN THK.=',F7.5,2X,'FIN SPAC.=',F6.4,2X,'FIN H.=',
     & F6.4,2X,'FINS=',F7.1,/)
235   FORMAT(10X,'ROWS=',F4.1,2X,'L. CKT=',F5.1,2X,'KFIN=',F6.2,
     & 1X,'BTU/HR*FT*F',////)
240   FORMAT(25X,'T H E   T U B E S   A R E   D R Y')
245   FORMAT(25X,'T H E   T U B E S   A R E   W E T')
250   FORMAT(/,20X,'QREF=',F7.1,/)
255   FORMAT(20X,'QSUM=',F7.1,/)
260   FORMAT(25X,'* * * PROGRAM TERMINATED * * *',2X,'JCR=',I1)
265   FORMAT(25X,'WARNING-LEAVING REFRIGERANT IS WET')
C
270   CONTINUE
      END
C
C****************************************
C THIS SUBROUTINE CALCULATES FLOW AREAS.
C****************************************
      SUBROUTINE FLAREA(DO,DELICE,PI,Y,FINS,HC,DC,LENGTH,
     & AFRONT,ATOTEL,ATBEL,AFINEL,AMINFL,DHEL,NELEM,DEQUIV,FHEQIV)
      REAL LENGTH,MFAEL,MF2PAL
      DIAO=DO+DELICE
      ATB=PI*DIAO*(LENGTH-(Y+2*DELICE)*FINS)
      AFIN=2.0*FINS*(HC*DC-PI*(DIAO**2.0)/4.0)
      ATOT=ATB+AFIN
      ATOTEL=ATOT/NELEM
      ATBEL=ATB/NELEM
      AFINEL=AFIN/NELEM
```

```
      AMINFL=(HC-DIAO)*(LENGTH-(Y+2*DELICE)*FINS)/NELEM
      DHEL=4*AMINFL*DC/ATOTEL
      AFRONT=HC*LENGTH
C
C THE RADIUS OF A CIRCULAR FIN OF SAME AREA AS THE FLAT PLATE FIN
C MODELLED IS CALCULATED.
C-----------------------------------------------------------------
      DEQUIV=2*(HC*DC/PI)**0.5
      FHEQIV=(DEQUIV-DO)/2.0
      RETURN
      END
C
C
C*****************************************************************
C THIS SUBROUTINE CALCULATES FROST DENSITY AND FROST THERMAL
C CONDUCTIVITY AS SUGGESTED BY MALHAMMAR (1986).
C*****************************************************************
      SUBROUTINE FROST2(I,JCR,NELEM,TMCOND,DELICE,RESICE,Q,QLAT,S,Y,
     & TWALL,ATL,VSPEC,DENSDA,MUA,DENICE,TIME,DTIME,ROWS,VFACE,INDVEL,
     & QRATS,DPQRS,REYNS,TWS)
      REAL TMCOND(50),DELICE(50),RESICE(50),Q(50),QLAT(50)
      REAL QRATS(50),DPQRS(50),REYNS(50),TWS(50)
      REAL MUA
      IF (QLAT(I) .LT. 0.02) RETURN
      TIME2=TIME+DTIME
      FTPREV=0
      KOUNT=0
      TWS(I)=TWS(I)+TWALL*DTIME
      TWAV=TWS(I)/TIME2
      TSURF=(TWAV-32.0)/1.8
      SO=S-Y
C
      SS=SO-2*DELICE(I)
      WSP=VFACE*S/(SS*60)
      IF (INDVEL .EQ. 1) WSP=VSPEC/60
C
      DPDTMP=4.325E+10*EXP(-5619/(273+TSURF))
      DPQR=DPDTMP*Q(I)/QLAT(I)
      DPQRS(I)=DPQRS(I)+DPQR*DTIME
      DPQRAV=DPQRS(I)/TIME2
C
      DEN2=4.98E-04*(1.0+TSURF/396.0)
      QRAT=QLAT(I)/Q(I)
```

```
      QRATS(I)=QRATS(I)+QRAT*DTIME
      QRATAV=QRATS(I)/TIME2
      TNUM=TIME2*3600*2.84E+06*QRATAV/DEN2
C
      REYNO=3600*WSP*2*SO*DENSDA/MUA
      REYNS(I)=REYNS(I)+REYNO*DTIME
      REYNM=REYNS(I)/TIME2
      REYN=0.5*(REYNM+REYNO)
      IF (REYN .LT. 2600) THEN
       VM=204
       SKI=2.58E-14+1.91E-16*REYN
       GOTO 10
      END IF
      IF (REYN .LT. 22000) THEN
       VM=113.0+3.50E-02*REYN
       SKI=5.23E-13
       IF (REYN .LT. 5000) GOTO 10
5      FT=VM*(1.05+(0.693+SKI*TNUM)**0.5)
       DENSFR=FT*DPQRAV/462.0
       IF (KOUNT .GT. 20) GOTO 15
       IF (ABS(FTPREV-FT) .LT. 0.5) GOTO 15
       KOUNT=KOUNT+1
       VM=295
       SKI=3.06E-20*FT*DENICE/DENSFR
       FTPREV=FT
       GOTO 5
      ELSE
       JCR=6
      END IF
10    FT=VM*(1.05+(0.693+SKI*TNUM)**0.5)
      DENSFR=FT*DPQRAV/462.0
15    IF (DENSFR .GT. 380) DENSFR=380
      IF (DENSFR .LT. 90) DENSFR=90
      CONDFR=0.202*DENSFR*(1.0-DENSFR/1860.0)/(VM-0.189*DENSFR)
C
      DENSFR=0.06248*DENSFR
      CONDFR=0.64316*CONDFR
      PROD=CONDFR*DENSFR
      DELICE(I)=TMCOND(I)/(ATL*DENSFR)
      RESICE(I)=DELICE(I)/(ATL*CONDFR*ROWS)
      RETURN
      END
C
```

```
C
C***********************************************************************
C THIS SUBROUTINE CALCULATES THE PRESSURE DROP ON THE AIR SIDE OF
C THE COIL AS A FUNCTION OF FROST ACCUMULTION ON THE COIL SURFACE.
C***********************************************************************
      SUBROUTINE PDROP(NELEM,DO,DELICE,PI,Y,FINS,HC,DC,LENGTH,AFRONT,
     & MF2PA,MUA,DENSDA,AFROST,MFA,S,DPAIR,ROWS,VFACE,HBAR)
      REAL DELICE(50)
      REAL LENGTH,MF2PA,MUA,MFA
      DICE=0
      DO 5 J=1,NELEM
    5 DICE=DICE+DELICE(J)
      HBAR=DICE/NELEM
      DEL=0
      CALL FLAREA(DO,DEL,PI,Y,FINS,HC,DC,LENGTH,
     & AFRONT,ATL,ATBEL,AFINEL,AMFL,DH,NELEM,DEQUIV,FHEQIV)
      SS1=S-Y-2*HBAR
      WSP=VFACE*S/(SS1*60)
      FR2=12.23*AFROST/(ATL*NELEM*ROWS)
      DPAIR2=FR2*DC*DENSDA*(WSP**2)/(2*SS1*2*32.2)
      DPAIR=12*DPAIR2/62.4
      RETURN
      END
C
C***********************************************************************
C THIS SUBROUTINE FINDS THE STATE OF THE AIR LEAVING AN ELEMENT.
C IT IS USED FOR DEHUMIDIFIED AND DRY AIR.
C***********************************************************************
      SUBROUTINE TLEAV2(I,KK2,HA2,C,TW,TA1,Z,HWALL,NELEM)
      DIMENSION TW(50),Z(9),TA1(50,9),STORE(9)
      K1=0
      K2=0
      IF (C .GT. 0) THEN
C
C     THE AIR IS DEHUMIDIFIED.
C---------------------------------------
      T2NEW=(0.5*TA1(I,1)+1.5*TW(I))/2
      Z(1)=TW(I)
      Z(3)=1.0
      CALL PSYC(Z,3)
      HWALL=Z(5)
      IF (HA2 .LT. Z(5)) GOTO 10
    5 W=TA1(I,4)-C*(TA1(I,1)-T2NEW)
```

```
      Z(1)=T2NEW
      Z(4)=W
      CALL PSYC(Z,4)
      DIFF=HA2-Z(5)
      IF (ABS(DIFF) .LT. 0.005) RETURN
      IF (DIFF .LT. 0) THEN
       K1=1
       T2RHS=T2NEW
       H2R=Z(5)
       T2NEW=T2RHS-1.0
       IF (T2NEW .LT. TW(I)) T2NEW=TW(I)
      ELSE
       K2=1
       T2LHS=T2NEW
       H2L=Z(5)
       T2NEW=T2LHS+1.0
       IF (T2NEW .GT. TA1(I,1)) T2NEW=TA1(I,1)
      END IF
      IF ((K1 .EQ. 1) .AND. (K2 .EQ. 1)) THEN
       T2NEW=T2RHS-(H2R-HA2)*(T2RHS-T2LHS)/(H2R-H2L)
      END IF
      GOTO 5
      ELSE
C
C  THE AIR IS DRY
C-------------------------
      T2=TA1(I,1)-2
      Z(1)=TW(I)
      Z(4)=TA1(I,4)
      CALL PSYC(Z,4)
      HWALL=Z(5)
      IF (HA2 .LT. Z(5)) GOTO 15
      CALL PROP1(HA2,TA1(I,4),T2,Z)
      END IF
      RETURN
C
   10 TOLER=0.005
      RH=1.0
      KSPEC=5
      TEV=TW(I)
      CALL PROP2(HA2,RH,TEV,Z,KSPEC,TOLER)
      RETURN
   15 RH=1.0
```

```
      W=TA1(I,4)
      KSPEC=4
      TOLER=1.0E-05
      CALL PROP2(W,RH,T2,Z,KSPEC,TOLER)
      IF (Z(5) .LT. HA2) THEN
       CALL PROP1(HA2,TA1(I,4),T2,Z)
       RETURN
      ELSE
       GOTO 10
      END IF
      END
C
C***********************************************************************
C THIS SUBROUTINE USES ENTHALPY OR SPECIFIC HUMIDITY AT SATURATION
C TO DETERMINE THE REMAINING PROPERTIES OF AIR.
C***********************************************************************
      SUBROUTINE PROP2(W,RH,T2,Z,KSPEC,TOLER)
      DIMENSION Z(9)
      KT=0
      J1=0
      J2=0
    5 TEV=T2
      KT=KT+1
      Z(1)=TEV
      Z(3)=RH
      CALL PSYC(Z,3)
      DIFF=Z(KSPEC)-W
      IF (KT .GT. 20) RETURN
      IF (ABS(DIFF) .LT. TOLER) RETURN
      IF (DIFF .LT. 0) THEN
       T2=TEV+2
       WLHS=Z(KSPEC)
       TLHS=TEV
       J1=1
      ELSE
       T2=TEV-2
       WRHS=Z(KSPEC)
       TRHS=TEV
       J2=1
      END IF
      IF ((J1 .EQ. 1) .AND. (J2 .EQ. 1)) THEN
       T2=TLHS+(W-WLHS)*(TRHS-TLHS)/(WRHS-WLHS)
      END IF
```

```
      GOTO 5
      END
C
C*****************************************************************
C THIS SUBROUTINE USES ENTHALPY AND SPECIFIC HUMIDITY TO DETERMINE
C THE REMAINING PROPERTIES OF THE AIR.
C*****************************************************************
      SUBROUTINE PROP1(H,W,T2,Z)
      DIMENSION Z(9)
      J1=0
      J2=0
      KT=0
    5 TEV=T2
      KT=KT+1
      Z(1)=TEV
      Z(4)=W
      CALL PSYC(Z,4)
      DIFF=Z(5)-H
      IF (KT .GT. 20) RETURN
      IF (ABS(DIFF) .LT. 0.005) RETURN
      IF (DIFF .LT. 0) THEN
       T2=TEV+2
       HLHS=Z(5)
       TLHS=TEV
       J1=1
      ELSE
       T2=TEV-2
       HRHS=Z(5)
       TRHS=TEV
       J2=1
      END IF
      IF ((J1 .EQ. 1) .AND. (J2 .EQ. 1)) THEN
       T2=TLHS+(H-HLHS)*(TRHS-TLHS)/(HRHS-HLHS)
      END IF
      GOTO 5
      END
C
C
C*****************************************************************
C                     SUBROUTINE REFLV
C THIS SUBROUTINE DETERMINES THE STATE AT THE END OF AN ELEMENT
C KNOWING THE LEAVING ENTHALPY AND SATURATION TEMP ,OR IN THE CASE
C OF SUPERHEATED REFRIGERANT, PRESSURE AT NODE 1.
```

```
C*********************************************************************
      SUBROUTINE REFLV(JCR,I,MF2PR,X,V,HR,PR,TR,NNODE,DI,XLONG)
      INTEGER I
      REAL MF2PR
      DIMENSION X(51),V(51),HR(51),PR(51),TR(51)
      JCR=0
      IF ((X(I) .LT. 1.0) .AND. (X(I) .GT. 0)) THEN
         CALL WETR(JCR,I,MF2PR,X,V,HR,PR,TR,NNODE,DI,XLONG)
      ELSE
         IF (X(I) .GE. 1.0) THEN
            CALL SUPREF(I,NNODE,TR,PR,HR,V,MF2PR,XLONG,DI)
            X(I+1)=1.0000001
         ELSE
            WRITE(20,5)
            JCR=1
         END IF
      END IF
    5 FORMAT(20X,'SUBCOOLED REFRIGERANT IS NOT CONSIDERED')
      RETURN
      END
C
C
C ********************************************************************
C                           SUBROUTINE WETR
C THIS SUBROTINE CALCULATES 2-PHASE REFRIGERANT STATE AT THE END
C OF AN ELEMENT.
C ********************************************************************
      SUBROUTINE WETR(JCR,I,MF2PR,X,V,HR,PR,TR,NNODE,DI,XLONG)
      REAL MF2PR,MURVS,MURLS,MUR
      DIMENSION X(51),V(51),HR(51),PR(51),TR(51)
      JCR=0
      K=0
      CALL SATPRP(ITR,TR(I),PSAT,VF,VG,HF,HFG,HG,SF,SG)
      MURLS=3600*VSC(TR(I),0)
      MURVS=3600*VSC(TR(I),1.0)
      X(I+1)=(HR(I+1)-HF)/HFG
    5 X2APX=X(I+1)
      XX=(X(I+1)+X(I))/2.0
      MUR=(1.0-XX)*MURLS+XX*MURVS
      REYN=MF2PR*DI/MUR
      CF=0.079*(1.0/REYN)**0.25
      DPRFR=-2.0*(MF2PR/3600)**2*CF*1.0/DI*(VF+XX*(VG-VF))*XLONG/
     & (144.0*32.2)
```

```
      V(I+1)=VF+X(I+1)*(VG-VF)
      DPRAC=-((MF2PR/3600)**2)*(V(I+1)-V(I))/(144*32.2)
      PR(I+1)=PR(I)+DPRAC+DPRFR
      TR(I+1)=TSAT(ITR,PR(I+1))
      CALL SATPRP(ITR,TR(I+1),PSAT,VF,VG,HF,HFG,HG,SF,SG)
      X(I+1)=(HR(I+1)-HF)/HFG
      K=K+1
      IF (K .GT. 20) GOTO 10
      IF (ABS(X(I+1)-X2APX) .GT. 0.0002) GOTO 5
   10 CONTINUE
      RETURN
      END
C
C*********************************************************************
C THIS SUBROUTINE CALCULATES THE CONDITION OF THE REFRIGERANT IN THE
C SUPERHEATED REGION. IT MAKES USE OF SUBROUTINE TVAP.
C*********************************************************************
      SUBROUTINE SUPREF(I,NNODE,TR,PR,HR,V,MF2PR,XLONG,DI)
      DIMENSION HR(51),TR(51),PR(51),V(51)
      REAL MUR,MF2PR
      K=0
      MUR=3600*VSC(TR(I),1.0)
      REYN=MF2PR*DI/MUR
      IF (REYN .LT. 10E+5) THEN
        FF=0.3164/REYN**0.25
      ELSE
        FF=0.0032+0.221/REYN**0.237
      END IF
      TMIN=TSAT(ITR,PR(I))
      TR(I+1)=TR(I)
      PR(I+1)=PR(I)
    5 TEST=TR(I+1)
      K=K+1
      CALL TVAP(ITR,PR(I+1),HR(I+1),TMIN,TR(I+1),V(I+1),S)
      DPFR=-FF*XLONG*V(I)*(MF2PR/3600)**2/(DI*2*32.2*144.0)
      PR(I+1)=PR(I)+DPFR
      IF (K .GT. 15) GOTO 10
      IF (ABS(TR(I+1)-TEST) .GT. 0.0005) GOTO 5
   10 RETURN
      END
C
C ******************************************************************
C                     SUBROUTINE TVAP
```

```
C THIS SUBROUTINE CALCULATES THE SUPERHEATED REFRIGERANT STATE AT
C THE END OF AN ELEMENT.
C *****************************************************************
      SUBROUTINE TVAP(ITR,P,H,TMIN,T,VR,S)
      K=0
    5 IF (T .LT. TMIN) T=TMIN
      CALL VAPOR(ITR,T,P,VR,HV,S)
      K=K+1
      DEL=ABS(H-HV)
      IF (DEL .LT. 0.0002) RETURN
      IF (K .GT. 16) RETURN
      IF (K .GE. 2) GOTO 10
      T1=T
      H1=HV
      IF (HV .GT. H) T=T-2
      IF (HV .LT. H) T=T+2
      GOTO 5
   10 T2=T
      H2=HV
      T=T2+(H-H2)*(T2-T1)/(H2-H1)
      T1=T2
      H1=H2
      GOTO 5
      END
C
C******************************************************************
C                          SUBROUTINE CALC
C THIS SUBROUTINE CALCULATES THE REFRIFERANT SIDE HEAT TRANSFER
C COEFFICIENT FOR TWO PHASE FLOW. THE CORRELATION USED IS THAT OF
C PIERRE (1955). A SUBROUTINE WHERE REFRIGERANT HEAT TRANSFER
C COEFFICIENT IS CALCULATED AS SUGGESTED BY TONG, IS INCLUDED BUT
C NOT ACTIVE. CAN BE DELETED. THE REFRIGERANT PROPERTIES IN
C TONG'S CORRELATION ARE EVALUATED AT, TEVAL=TREF+0.33*(TW-TREF).
C******************************************************************
      SUBROUTINE CALC(JCR,I,MF2PR,DI,AIEL,TEVAL,DXDL,DQDEL,X,V,HR,
     & PR,TR,HTR,RESR,NNODE,SS,ROWS,PI,XO)
      INTEGER JCR,I
      DIMENSION X(51),V(51),HR(51),PR(51)
      DIMENSION TR(51),SS(51)
      REAL KRLS,MURLS,MUR,MF2PR,MURVS,KR
C------------------------------------------------------------------
C IN PIERRE'S CORRELATION THE PROPERTIES ARE EVALUATED AT TREF.
C-----------------------------------------------------------------
```

```
      TEVAL=TR(I)
C-----------------------------------------------------------------
        IF (X(I) .LT. 1) THEN
          CALL SATPRP(ITR,TEVAL,PSAT,VF,VG,HF,HFG,HG,SF,SG)
          RORLS=1/VF
          RORVS=1/VG
          MURLS=VSC(TEVAL,0)
          MURVS=VSC(TEVAL,1.0)
          MURLS=3600*MURLS
          MURVS=3600*MURVS
          KRLS=COND(TEVAL,0)
          CPRLS=SH(TEVAL,0)
C
C THE 2 PHASE HEAT TRANSFER COEFFICIENT AS SUGGESTED BY TONG IS
C CALCULATED IN HTR2.
C------------------------------------------------------------------------
C         CALL HTR2(JCR,I,MF2PR,KRLS,DI,MURLS,MURVS,X,CPRLS,RORLS,
C    &    RORVS,HTR)
C
C THE 2 PHASE REFRIGERANT HEAT TRANSFER COEFFICIENT AS SUGGESTED
C BY BO PIERRE IS CALCULATED IN SUBROUTINE HTRDOM.
C------------------------------------------------------------------------
          CALL SATPRP(ITR,TR(I),PR(I),VF,VG,HF,HFG,HG,SF,SG)
          CALL HTRDOM(JCR,I,MF2PR,KRLS,MURLS,DXDL,DI,HFG,SS,
     &    X,HTR,NNODE)
          IF (X(I) .GT. XO) THEN
           CALL SPHT(ITR,TEVAL,PR(I),SV,CPR,GAMA,SONIC)
           KR=COND(TEVAL,1.0)
           CALL HTR1(JCR,MF2PR,CPR,KR,MURVS,DI,HTR1PH)
           WGT=(1.0-X(I))/(1.0-XO)
           HTR=HTR*(SIN(PI*WGT/2))**2+HTR1PH*(COS(PI*WGT/2))**2
          END IF
C-----------------------------------------------------------------
          HR(I)=HF+X(I)*HFG
          V(I)=VF+X(I)*(VG-VF)
        ELSE
          CALL VAPOR(ITR,TR(I),PR(I),VVAP,HVAP,SVAP)
          HR(I)=HVAP
          V(I)=VVAP
          CALL SPHT(ITR,TR(I),PR(I),SV,CPR,GAMA,SONIC)
          KR=COND(TR(I),1.0)
          MUR=VSC(TR(I),1.0)
          MUR=3600*MUR
```

```
         CALL HTR1(JCR,MF2PR,CPR,KR,MUR,DI,HTR)
      END IF
      RESR=1/(HTR*AIEL*ROWS)
      RETURN
      END
C
C ***********************************************************************
C                            SUBROUTINE HTR2
C  THIS SUBROUTINE CALCULATES THE REFRIGERANT 2-PHASE HEAT
C  TRANSFER COEFFICIENT AS DESCRIBED BY TONG IN "BOILING HEAT
C  TRANSFER AND TWO PHASE FLOW", P.122.
C ***********************************************************************
      SUBROUTINE HTR2(JCR,I,MF2PR,KRLS,DI,MURLS,MURVS,X,CPRLS,RORLS,
     & RORVS,HTR,NNODE)
      INTEGER JCR,I
      REAL MF2PR,KRLS,MURLS,MURVS,MALP,NN
      DIMENSION X(51)
      AA=2.5
      NN=0.75
      HTLR=0.023*(KRLS/DI)*((DI*MF2PR*(1.0-X(I))/MURLS)
     & **0.8)*((CPRLS*MURLS/KRLS)**0.4)
      MALP=((X(I)/(1.0-X(I)))**0.9)*((RORLS/RORVS)**0.5)*
     & ((MURVS/MURLS)**0.1)
      IF (MALP .LT. 3) THEN
         WRITE(30,5)
         JCR=3
      END IF
      HTR=AA*HTLR*(MALP**NN)
    5 FORMAT(25X,'* * * MALP IS LESS THAN 3 * * *')
      RETURN
      END
C
C
C***********************************************************************
C THIS SUBROUTINE CALCULATES THE 2-PHASE REFRIGERANT HEAT TRANSFER
C COEFFICIENT. IT USES THE SEMI EMPIRICAL CORRELATION SUGGESTED BY
C BO PIERRE, APPLICABLE FOR MASS FLOWS AND HEAT FLUX DENSITIES IN THE
C REGION OF INTEREST.
C***********************************************************************
      SUBROUTINE HTRDOM(JCR,I,MF2PR,KRLS,MURLS,DXDL,DI,HFGR,SS,
     & X,HTR,NNODE)
      DIMENSION SS(51),X(51)
      REAL MF2PR,KRLS,MURLS,KF
```

```
      RE=MF2PR*DI/MURLS
      KF=778*DXDL*HFGR/(SS(I+1)-SS(I))
      PARA=RE*RE*KF
      IF (PARA .GT. 0.7E+12) JCR=4
      IF (PARA .LT. 1E+09) JCR=5
      HTR=0.0009*(KRLS/DI)*PARA**0.5
      RETURN
      END
C
C
C ****************************************************************************
C                          SUBROUTINE HTR1
C  THIS SUBROUTINE CALCULATES THE HEAT TRANSFER COEFFICIENT ON THE
C  SUPERHEATED REFRIGERANT SIDE. THE CORRELATION IS SUGGESTED BY
C  HILLER (1976).
C ****************************************************************************
      SUBROUTINE HTR1(JCR,MF2PR,CPR,KR,MUR,DI,HTR)
      REAL MF2PR,MUR,KR
      RER=MF2PR*DI/MUR
      HTR=0.0108*MF2PR*(CPR**(0.333))*(KR**(0.667))*((1.0/MUR)**
     & 0.667)*((1.0/RER)**0.1375)
      RETURN
      END
C
C ****************************************************************************
C                          SUBROUTINE HEATC
C THIS SUBROUTINE CALCULATES THE AIR SIDE HEAT TRANSFER COEFFICIENT.
C THE CORRELATION IS THAT OF BRIGGS AND YOUNG (1962). THIS CORRELATION
C IS PRESENTLY NOT IN USE.
C ****************************************************************************
      SUBROUTINE HEATC(KA,CPA,PRA,MUA,MF2PA,DO,S,FH,Y,HTA)
      REAL KA,MUA,MF2PA
      HTA=0.134*KA/DO*((MF2PA*DO/MUA)**0.681)*(PRA**0.333)*
     & ((S/FH)**0.2)*((S/Y)**0.1134)
      RETURN
      END
C
C
C*************************************************************************
C THIS SUBROUTINE CALCULATES THE HEAT TRANSFER COEFFICIENT AS
C SUGGESTED BY MCQUISTON (1983).
C*************************************************************************
      SUBROUTINE HEAT2(CPA,PRA,MF2PA,MUA,DO,ASAT,HTA,DC)
```

```
      REAL MF2PA,MUA
      REA=MF2PA*DO/MUA
      REB=MF2PA*DC/MUA
      RFAC=(1.0/REB)**1.2
      SJP=(1.0/REA)**0.4*(1.0/ASAT)**0.15
      SJ4=0.2675*SJP+1.325E-03
      SJ=((1.0-1280*RFAC)/(1.0-5120*RFAC))*SJ4
      HTA=MF2PA*CPA*SJ/PRA**0.667
      RETURN
      END
C
C**********************************************************************
C THIS SUBROUTINE CALCULATES THE FIN EFFICIENCY OF A PLATE FIN HEAT
C EXCHANGER AS SUGGESTED BY GARDNER(1945). THE CURVE FIT EQUATIONS ARE
C THOSE OF TANTAKITTI(1985).
C**********************************************************************
      SUBROUTINE FINEF3(HTA,KFIN,HFGW,Y,DO,DEQUIV,FHEQIV,NF)
      REAL KFIN,NF
      R=DEQUIV/DO
      X=FHEQIV*(2.0*HTA/(KFIN*Y))**0.5
      PHI1=1.03552448-0.31837607*X+0.02589744*X*X-0.00101*X**3
      PHI15=1.04188811-0.3713209*X+0.04282051*X*X-0.000637*X**3
      PHI2=1.03552448-0.45592852*X+0.08051282*X*X-0.005004*X**3
      PHI3=1.03881119-0.5195338*X+0.10275058*X*X-0.0070863*X**3
      PHI4=1.03230769-0.5540948*X+0.117016*X*X-0.0087024*X**3
      IF (R .LE. 1.0) THEN
        NF=PHI1
        GOTO 5
      END IF
      IF (R .LE. 1.5) THEN
        NF=2.0*(PHI15-PHI1)*(R-1.0)+PHI1
        GOTO 5
      END IF
      IF (R .LE. 2.0) THEN
        NF=2.0*(PHI2-PHI15)*(R-1.5)+PHI15
        GOTO 5
      END IF
      IF (R .LE. 3.0) THEN
        NF=(PHI3-PHI2)*(R-2.0)+PHI2
        GOTO 5
      END IF
      IF (R .LE. 4.0) THEN
        NF=(PHI4-PHI3)*(R-3.0)+PHI3
```

```
         GOTO 5
        END IF
        NF=PHI4
    5   CONTINUE
        RETURN
        END
C
C *********************************************************************
C                          SUBROUTINE RSLTS
C THIS SUBROUTINE TAKES ALL VECTORS HOLDING THE RESULTS AND WRITES
C THEM TO AN OUTPUT FILE.
C *********************************************************************
        SUBROUTINE RSLTS(TA1,TA2,Q,TW,TR,PR,X,HTAIR,HTREF,SS,HR,
     &  EFF,CONDEL,RESIST,DELICE,VLOC,NNODE,NELEM,VREF,TFR)
        DIMENSION TA1(50,9),TA2(50,9)
        DIMENSION Q(50),TR(51),PR(51),X(51),SS(51)
        DIMENSION HTAIR(50),HTREF(50),EFF(50),TW(50)
        DIMENSION HR(51),CONDEL(50),VREF(51),TFR(50)
        DIMENSION RESIST(50,3),DELICE(50),RESICE(50),VLOC(50)
        REAL MFR
        ICTR=1
        WRITE(20,40)
        WRITE(20,45)
        WRITE(20,55)
        DO 5 J=1,NELEM
    5   WRITE(20,50) J,SS(J),(TA1(J,I),I=1,9)
        WRITE(20,*)
        WRITE(20,120) ICTR
        WRITE(20,35)
        WRITE(20,45)
        WRITE(20,55)
        DO 10 J=1,NELEM
   10   WRITE(20,50) J,SS(J),(TA2(J,I),I=1,9)
        WRITE(20,*)
        WRITE(20,120) ICTR
        WRITE(20,60)
        WRITE(20,45)
        WRITE(20,65)
        DO 15 J=1,NNODE
   15   WRITE(20,70) SS(J),X(J),TR(J),PR(J),HR(J),VREF(J)
        WRITE(20,*)
        WRITE(20,120) ICTR
        WRITE(20,75)
```

```
      WRITE(20,45)
      WRITE(20,80)
      DO 20 J=1,NELEM
20    WRITE(20,85) SS(J),J,HTAIR(J),HTREF(J),EFF(J),Q(J)
      WRITE(20,*) ' '
      WRITE(20,120) ICTR
      WRITE(20,90)
      WRITE(20,45)
      WRITE(20,95)
      DO 25 J=1,NELEM
25    WRITE(20,100) SS(J),J,TW(J),TFR(J),CONDEL(J)
      WRITE(20,*) ' '
      WRITE(20,120) ICTR
      WRITE(20,105)
      WRITE(20,45)
      WRITE(20,110)
      DO 30 J=1,NELEM
      DLICE=12.0*DELICE(J)
30    WRITE(20,115) SS(J),J,RESIST(J,1),RESIST(J,2),RESIST(J,3),
     & DLICE,VLOC(J)
      WRITE(20,*) ' '
35    FORMAT(20X,'L E A V I N G   A I R   T E M P E R A T U R E S')
40    FORMAT(20X,'E N T E R I N G  A I R  T E M P E R A T U R E S')
45    FORMAT(19X,'---------------------------------------------',
     & '---',/)
50    FORMAT(6X,I3,2X,F5.2,2X,F5.2,2X,F5.2,1X,F5.3,2X,E8.2,2X,F4.1,
     & 2X,F4.1,2X,F4.1,2X,F4.2,2X,F4.1)
55    FORMAT(4X,'ELEMENT',1X,'DIST',2X,'DR.B.',2X,'WT.B.',2X,'R.H.',
     & 3X,'SP.HU.',3X,'ENTH',2X,'D.PT',2X,'S.H.',2X,'L.H.',2X,'S.V.')
60    FORMAT(20X,'R E F R I G E R A N T   T E M P E R A T U R E S')
65    FORMAT(4X,'DISTANCE',2X,'QUALITY',2X,'  TEMP.  ',2X,'  PRES.  ',
     & 2X,'  ENTH.  ',2X,' REF.VEL. ')
70    FORMAT(5X,F5.2,5X,F5.3,4X,F6.2,4X,F5.2,5X,F6.2,4X,F7.1)
75    FORMAT(20X,'HEAT TRANSFER COEFFICIENTS')
80    FORMAT(4X,'DISTANCE',2X,'ELEMENT',2X,'H.T.C.AIR',2X,'H.T.C.REF',
     & 2X,'FINEF',3X,'Q.ELE.')
85    FORMAT(5X,F5.2,6X,I3,6X,F6.2,4X,F7.1,3X,F5.3,3X,F7.2)
90    FORMAT(20X,'CONDENSATION RATES AND WALL TEMPERATURES')
95    FORMAT(4X,'DISTANCE',2X,'ELEMENT',2X,'TEMP.WALL',2X,'TEMP.FST',
     & 2X,'COND.RATE')
100   FORMAT(5X,F5.2,6X,I3,6X,F6.2,4X,F6.2,4X,F7.3)
105   FORMAT(27X,'R E S I S T A N C E   V A L U E S')
110   FORMAT(4X,'DISTANCE',2X,'ELEMENT',2X,'  RESA  ',2X,' RESFST',2X,
```

```
     & '  RESR ',2X,'FST THK',2X,' VLOCAL')
 115  FORMAT(5X,F5.2,6X,I3,4X,F7.5,2X,F7.5,2X,F7.5,2X,F7.5,2X,F7.2)
 120  FORMAT(I1)
      RETURN
      END
C
C
C*********************************************************************
C  IN THIS SUBROUTINE OVERALL HEAT TRANSFER RATES AND AVERAGE
C  TEMPERATURES ARE CALCULATED.
C*********************************************************************
      SUBROUTINE OVERAL(MFAEL,NELEM,TA1,TA2,TW,CONDEL,TIME,DELICE,
     & PRNL,X,JFLAG,MFAET,MFA,ACFROST,INDIC,PERC,MFR,TFR,HFGW)
      REAL MFAEL,MFA,MFR
      DIMENSION TA1(50,9),TA2(50,9),CONDEL(50),TW(50),DELICE(50)
      DIMENSION Z(9),X(50),TFR(50)
      REAL MFAET(50)
      ICTR=1
C
C  TOTAL HEAT TRANSFER RATES ARE CALCULATED.
C--------------------------------------------------------
      AMF2PH=0
      AMFSUP=0
      K2P=0
      KSUP=0
      AWL2PH=0
      HTOT2P=0
      CONDT=0
      QLAT=0
      QSEN=0
      QTOT=0
      HLAT2=0
      HSEN2=0
      HTOT2=0
      SHUM2=0
      DO 5 J=1,NELEM
      IF ((X(J)-0.9999) .LE. 0) THEN
       HLAT2=HLAT2+TA2(J,8)*MFAET(J)
       HSEN2=HSEN2+TA2(J,7)*MFAET(J)
       SHUM2=SHUM2+TA2(J,4)*MFAET(J)
       HTOT2=HTOT2+TA2(J,5)*MFAET(J)
       AMF2PH=AMF2PH+MFAET(J)
       K2P=K2P+1
```

```
      ELSE
       HLAT1=HLAT1+TA2(J,8)*MFAET(J)
       HSEN1=HSEN1+TA2(J,7)*MFAET(J)
       SHUM1=SHUM1+TA2(J,4)*MFAET(J)
       HTOT1=HTOT1+TA2(J,5)*MFAET(J)
       AMFSUP=AMFSUP+MFAET(J)
       KSUP=KSUP+1
      END IF
      CONDT=CONDT+CONDEL(J)
    5 QTOT=QTOT+MFAET(J)*(TA1(J,5)-TA2(J,5))
      QLAT=CONDT*HFGW
      QSEN=QTOT-QLAT
C
C     AVERAGE LEAVING ENTHALPIES AND HUMIDITIES ARE CALCULATED IN
C     REFR. 2 PHASE AND SUPERHEAT REGION.
C-----------------------------------------------------------------------
      AMFLOW=AMF2PH+AMFSUP
      AHLAT=(HLAT1+HLAT2)/AMFLOW
      AHSEN=(HSEN1+HSEN2)/AMFLOW
      ASHUM=(SHUM2+SHUM1)/AMFLOW
      AHTOT=(HTOT1+HTOT2)/AMFLOW
      T2=(TA2(1,1)+TA2(50,1))/2
      CALL PROP1(AHTOT,ASHUM,T2,Z)
      ADB=Z(1)
      AWB=Z(2)
      ARH=100*Z(3)
C
      HLAT2=HLAT2/AMF2PH
      HSEN2=HSEN2/AMF2PH
      HTOT2=HTOT2/AMF2PH
      SHUM2=SHUM2/AMF2PH
      TEV=TA2(1,1)
      CALL PROP1(HTOT2,SHUM2,TEV,Z)
      ADBWET=Z(1)
      AWBWET=Z(2)
      ARHWET=100*Z(3)
C
      IF (KSUP .GT. 0) THEN
       HLAT1=HLAT1/AMFSUP
       HSEN1=HSEN1/AMFSUP
       HTOT1=HTOT1/AMFSUP
       SHUM1=SHUM1/AMFSUP
       TS=TA2(NELEM,1)
```

```
      CALL PROP1(HTOT1,SHUM1,TS,Z)
      ADBSUP=Z(1)
      AWBSUP=Z(2)
      ARHSUP=100*Z(3)
     END IF
C
C  AVERAGE TEMPERATURES ARE CALCULATED
C----------------------------------------------
     DICE=0
     SFRWET=0
     SFRSUP=0
     SWLWET=0
     SWLSUP=0
     DO 10 J=1,NELEM
     IF ((X(J)-0.9999) .LE. 0) THEN
       SWLWET=SWLWET+TW(J)
       SFRWET=SFRWET+TFR(J)
       DICE=DICE+DELICE(J)
     ELSE
       SWLSUP=SWLSUP+TW(J)
       SFRSUP=SFRSUP+TFR(J)
     END IF
 10  CONTINUE
     AFRWET=SFRWET/K2P
     AWLWET=SWLWET/K2P
     DICE=12.0*DICE/K2P
     IF (KSUP .GT. 0) THEN
      AFRSUP=SFRSUP/KSUP
      AWLSUP=SWLSUP/KSUP
     END IF
     AFR=(SFRSUP+SFRWET)/NELEM
     AWL=(SWLSUP+SWLWET)/NELEM
C
C  THE CALCULATED VALUES ARE SENT TO THE OUTPUT FILE.
C-----------------------------------------------------------------
     WRITE(20,15) TIME
     WRITE(20,30)
     WRITE(20,35)
C
     WRITE(20,*)
     WRITE(20,145)
     WRITE(20,25)
     WRITE(20,150) PERC
```

```
      WRITE(20,140) MFA,MFR
C
      WRITE(20,*)
      IF ((CONDT .GT. 0) .AND. (TW(1) .LT. 32)) THEN
       WRITE(20,130)
       WRITE(20,45)
       WRITE(20,135) DICE,ACFROST
      END IF
      WRITE(20,40)
      WRITE(20,45)
      WRITE(20,50) QTOT,QSEN
      WRITE(20,55) QLAT
      WRITE(20,60)
      WRITE(20,45)
      WRITE(20,65) CONDT
C
      WRITE(20,75)
      WRITE(20,70)
      WRITE(20,90) ADBWET
      WRITE(20,95) AWBWET
      WRITE(20,100) AWLWET
      WRITE(20,105) ARHWET
      WRITE(20,110) SHUM2
      WRITE(20,115) HLAT2
      WRITE(20,120) HSEN2
      WRITE(20,125) HTOT2
C
      IF (KSUP .GT. 0) THEN
       WRITE(20,80)
       WRITE(20,70)
       WRITE(20,90) ADBSUP
       WRITE(20,95) AWBSUP
       WRITE(20,100) AWLSUP
       WRITE(20,105) ARHSUP
       WRITE(20,110) SHUM1
       WRITE(20,115) HLAT1
       WRITE(20,120) HSEN1
       WRITE(20,125) HTOT1
      END IF
C
      WRITE(20,85)
      WRITE(20,70)
      WRITE(20,90) ADB
```

```
      WRITE(20,95) AWB
      WRITE(20,100) AWL
      WRITE(20,105) ARH
      WRITE(20,110) ASHUM
      WRITE(20,115) AHLAT
      WRITE(20,120) AHSEN
      WRITE(20,125) AHTOT
C
      WRITE(20,155) ICTR
   15 FORMAT(33X,'TIME=',F5.2,/)
   20 FORMAT(33X,'--------------')
   25 FORMAT(23X,'-------------------------------------------------')
   30 FORMAT(25X,'CALCULATED OVERALL, AND AVERAGE RESULTS')
   35 FORMAT(22X,'==================================================')
   40 FORMAT(25X,'HEAT TRANSFER RATES (BTU/HR)')
   45 FORMAT(23X,'-----------------------------------')
   50 FORMAT(21X,'QTOTAL=',F7.1,4X,'QSENSIBLE=',F7.1)
   55 FORMAT(34X,'QLATENT=',F7.1,/)
   60 FORMAT(25X,'CONDENSATION RATE (LBM/HR)')
   65 FORMAT(23X,'TOTAL CONDENSATION RATE=',F6.3,//)
   70 FORMAT(23X,'-------------------------------------------')
   75 FORMAT(25X,'AVG.QTYS.IN THE 2 PHASE REF.REGION')
   80 FORMAT(25X,'AVG.QTYS.IN THE REF.SUPERHEAT REGION')
   85 FORMAT(25X,'AVG.QTYS. OVER LENGTH OF COIL')
   90 FORMAT(20X,'LEAVING DRY BULB =',F6.2,' F')
   95 FORMAT(20X,'LEAVING WET BULB =',F6.2,' F')
  100 FORMAT(20X,'WALL TEMPERATURE =',F6.2,' F')
  105 FORMAT(20X,'REL. HUMIDITY    =',F6.1,' %')
  110 FORMAT(20X,'LEAVING HUMIDITY =',E12.4,' LBWV/LBDA')
  115 FORMAT(20X,'LATENT ENTHALPY  =',F6.2,' BTU/LBM')
  120 FORMAT(20X,'SENSIBLE ENTHALPY=',F6.2,' BTU/LBM')
  125 FORMAT(20X,'TOTAL ENTHALPY   =',F6.2,' BTU/LBM',/)
  130 FORMAT(20X,'AVG FROST THK IN 2-PHASE REF.REGION.')
  135 FORMAT(15X,'AVERAGE THICKNESS=',F8.6,' INCHES',3X,
     & 'ACC.FROST=',F8.3,' LBM',/)
  140 FORMAT(20X,'AIR FLOW=',F7.1,' LBM/HR',3X,'REF.FLOW=',F7.1,
     & 'LBM/HR',/)
  145 FORMAT(25X,'PERCENT OF COIL IN 2 PHASE REF.REGION')
  150 FORMAT(33X,'SATURATED REGION=',F6.1,' %',/)
  155 FORMAT(I1)
      RETURN
      END
```

```
C*********************************************************************
C
C NUMERICAL MODELLING OF DIRECT EXPANSION HEAT EXCHANGER COILS.
C
C*********************************************************************
C
C THIS PROGRAM IMPLEMENTS THE THREE REGION MODEL. THE UNDERLYNG
C ASSUMPTIONS AND AN OUTLINE OF THE NUMERICAL PROCEDURE IS PART OF
C THE REPORT.
C
C     VARIABLE DEFINITIONS
C--------------------------------------------
C ADP.......APPARATUS DEW POINT
C AH........SATURATED ENTHALPY AY ADP.
C AW........SATURATED SPECIFIC HUMIDITY AT ADP.
C AFC.......FRACTION OF COIL IN REGION UNDER CONSIDERATION
C ATB.......TUBE SURFACE AREA EXPOSED TO AIR
C AFIN......FIN SURFACE AREA
C AFROST....ACCUMULATED FROST
C AI........INSIDE SURFACE AREA
C AMF.......MINIMUM FREE FLOW THROUGH AREA
C ARFC......FACE AREA
C ATOT......TOTAL AIR SIDE SURFACE AREA
C C.........SLOPE OF THE PSYCROMETRIC PROCESS LINE
C CC2PH1....COIL CHARACTERISTIC IN SAT'D REGION BASED ON INLET ENTH.
C CC2PHA....COIL CHARACTERISTIC IN SAT'D REGION BASED ON AVG. ENTH.
C CCOV1.....COIL CHARACTERISTIC BASE ON INLET TEMPERATURE
C CCOVA.....COIL CHARACTERISTIC BASED ON AVERAGE TEMPERATURE
C CFM.......VOLUMETRIC FLOW RATE (CUBIC FEET PER MINUTE)
C CPA.......SPECIFIC HEAT OF AIR
C CKTS......NUMBER OF CIRCUITS
C COND......CONDENSATION RATE
C CONDFR....FROST THERMAL CONDUCTIVITY
C CPAD......SPECIFIC HEAT DRY AIR
C CPR.......SPECIFIC HEAT OF REFRIGERANT
C DC........DEPTH OF ONE ROW (LONGITUDINAL SPACING)
C DELHD.....FROST LAYER THICKNESS AT PREVIOUS TIME STEP
C DELICE....FROST THICKNESS
C DENSA1....DENSITY OF INLET AIR
C DENSDA....DENSITY OF DRY AIR
C DENICE....FROST DENSITY
C DEQUIV....EQUIVALENT FIN DIAMETER
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C DI........INSIDE DIAMETER OF TUBE
C DH........HYDRAULIC DIAMETER
C DO........OUTSIDE DIAMETER OF TUBES
C DPAIR.....AIR SIDE PRESSURE COIL
C DPINIT....AIR SIDE PRESSURE DROP, NON FROSTED COIL
C DPTH......DEPTH OF COIL
C DTIME.....TIME STEP
C DTLM......LOG MEAN TEMPERATURE DIFFERENCE
C DTSHSP....SPECIFIED DEGREE OF SUPERHEAT
C E2PH1.....COIL ENTH. EFF. IN SAT'D REGION BASED ON INLET ENTH.
C E2PHA.....COIL ENTH. EFF. IN SAT'D REGION BASED ON AVG. ENTH.
C ESUP1.....COIL ENTH. EFF. IN SUPERHEATED REGION BASED ON INLET ENTH.
C ESUPA.....COIL ENTH. EFF. IN SUPERHEATED REGION BASED ON AVG. ENTH.
C EHOV1.....COIL ENTHALPY EFFECTIVENESS BASED ON INLET ENTHALPY
C EHOVA.....COIL ENTHALPY EFFECTIVENESS BASED ON AVERAGE ENTHALPY
C ETEMP1....COIL TEMPERATURE EFFECTIVENESS BASED ON INLET TEMP.
C ETEMPA....COIL TEMPERATURE EFFECTIVENESS BASED ON AVG.TEMP.
C FH........FIN HEIGHT
C FHEQIV....EQUIVALENT FIN HEIGHT
C FINS......NUMBER OF FINS
C FPI.......FIN DENSITY, FIN PER INCH
C HA1.......INLET AIR ENTHALPY
C HA2.......LEAVING AIR ENTHALPY
C HC........HEIGHT OF ONE ROW (LATITUDINAL SPACING)
C HCONT.....CONTACT CONDUCTANCE
C HFOULO....OUTSIDE FOULING CONTUCTANCE (AIR SIDE)
C HFOULI....INSIDE FOULING CONDUCTANCE (REF.SIDE)
C HFGW......HEAT OF VAPORIZATION, OR DESUBLIMATION
C HR1.......INLET REFRIGERANT ENTHALPY
C HT........HEIGHT OF COIL
C HTA.......AIR SIDE HEAT TRANSFERR COEFFICIENT
C HTR.......REFRIGERANT SIDE HEAT TRANSFER COEFFICIENT
C HWALL.....AIR ENTHALPY CORRESPONDING TO WALL TEMP.
C IFROST....FLAG. IF 1, THEN SCHEDULE OF INLET CONDITIONS
C INDVEL....FLAG. IF 1, THEN LOCAL AIR VELOCITY IS SPECIFIED
C INDIC.....FLAG. IF 1, SUPERHEAT SPEC'D, IF 2, OUTLET QUALITY SPEC'D.
C INDQL.....FLAG. IF 1 INLET QUALITY SPECIFIED
C INDTEV....FLAG. IF 1 EVAPORATOR TEMP. SPECIFIED
C INDVFC....FLAG. IF 1, FACE VELOCITY SPECIFIED
C JFLAG.....INDICATES COIL SURFACE CONDITION:
C             =1, CONDENSATE ON COIL SURFACE
C             =2, FROST ON COIL SURFACE
C KFIN......THERMAL CONDUCTIVITY OF FIN
```

```
C KFLAG.....IF 1, EVAPORATOR SMALLER THAN REQ'D FOR HEAT TRANSFER
C KA........THERMAL CONDUCTIVITY OF AIR
C KR........REFRIGERANT THERMAL CONDUCTIVITY
C KRLS......SATURATED LIQUID REFRIGERANT CONDUCTIVITY
C LENGTH....LENGTH OF CIRCUIT
C LSUP......LENGTH IN SUPERHEATED REGION
C LEN2PH....LENGTH IN TWO PHASE REGION
C LE........LEWIS NUMBER
C MFA.......AIR MASS FLOW RATE
C MFA2PH....AIR MASS FLOW IN TWO PHASE REGION
C MFASUP....AIR MASS FLOW IN SUPERHEATED REGION
C MFR.......REFRIGERANT MASS FLOW RATE PER CIRCUIT
C MFRT......TOTAL REFRIGERANT MASS FLOW RATE
C MF2PA.....AIR MASS FLUX
C MF2PR.....REFRIGERANT MASS FLUX
C MFROST....ACCUMULATED FROST
C MUA.......AIR DYNAMIC VISCOSITY
C MUR.......REFRIGERANT SIDE DYNAMIC VISCOSITY
C MURLS.....REFRIGERANT SATURATED LIQUID DYNAMIC VISCOSITY
C MURVS.....REFRIGERANT SATURATED VAPOR DYNAMIC VISCOSITY
C NF........FIN EFFICIENCY
C NTU.......NUMBER OF TRANSFER UNITS
C NDTPTS....NUMBER OF OPERATING POINTS
C NREF......REFRIGERANT NUMBER
C PERC......PERCENT OF COIL IN SATURATED REGION
C PI........PI, 3.14159....
C PINTVL....PRINTING INTERVAL
C PRA.......AIR PRANDTL NUMBER
C PR1.......INLET EVAP.TEMP (SATURATION)
C QLTYSP....SPECIFIED OUTLET QUALITY
C QUAL......INLET QUALITY
C Q.........HEAT TRANSFERRED
C QLAT......LATENT HEAT TRANSFERRED
C Q2PH......HEAT TRANSFERRED IN THE SAT'D REFRIGERANT REGION
C QSUP......HEAT TRANSFERRED IN SUPERHEATED REFRIGERANT REGION
C QTOT......TOTAL HEAT TRANSFERRED
C REA.......AIR SIDE REYNOLDS NUMBER
C RER.......REFRIGERANT SIDE REYNOLDS NUMBER
C RESCON....CONTACT RESISTANCE
C RESFLI....INSIDE FOULING RESISTANCE (REF.SIDE)
C RESFLO....OUTSIDE FOULING RESISTANCE (AIR SIDE)
C RESFM.....AIR SIDE THERMAL RESISTANCE NOT INCLUDING FIN EFFICIENCY
C RESI......FROST RESISTANCE MULTIPLIED BY AREA
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```
C RESICE....FROST THERMAL RESISTANCE
C RESR......REFRIGERANT SIDE THERMAL RESISTANCE
C RESA......AIR SIDE THERMAL RESISTANCE
C RH1.......AIR INLET RELATIVE HUMIDITY
C RH2.......AIR LEAVING RELATIVE HUMIDITY
C RH2MIX....RELATIVE HUM. OF MIXED AIR STREAM (SAT'D AND SUPERHEATED)
C RI1HD.....FROST THERMAL RES. MULT BY AREA AT PREVIOUS TIME STEP
C ROWS......NUMBER OF ROWS DEEP (CIRCUITS)
C RWSH......ROWS HIGH (PASSES PER CIRCUIT, 1 CIRCUIT= 1 ROW)
C S.........INTERSPATIAL FIN DISTANCE
C SUMR......SUMMATION OF THERMAL RESISTANCES
C TA1.......INLET AIR TEMPERATURE
C TA2.......LEAVING AIR TEMPERATURE
C TAV.......AVERAGE (LOG MEAN ) TEMPERATURE
C TA2MIX....MIXED LEAVING AIR TEMP. (SAT'D AND SUPERHEATED)
C TCO1HD....TOTAL DEHUM'N IN TWO PHASE REGION AT PREVIOUS TIME STEP
C TCO2HD....TOTAL DEHUM'N IN TRANSITION REGION AT PREVIOUS TIME STEP
C TFR.......COIL SURFACE TEMPERATURE (FROST, CONDENSATE OR COIL)
C TR1.......INLET REFRIGERANT TEMP.
C TLIMIT....SPECIFIED TIME LIMIT
C TMCOND....TOTAL DEHUMIDIFICATION IN A REGION OF THE COIL
C TOLER.....TOLERANCE
C TW........MEAN WALL TEMPERATURE
C VFCE......FACE VELOCITY
C WA1.......INLET SPECIFIC HUMIDITY
C WA2.......LEAVING SPECIFIC HUMIDITY
C WAV.......AVERAGE SPECIFIC HUMIDITY (CORR. TO LOG MEAN TEMP.)
C WA2MIX....MIXED SPECIFIC HUM. (SAT'D AND SUPERHEATED)
C WDTH......WIDTH OF COIL
C X1........INLET QUALITY
C X2........LEAVING QUALITY
C XSET......QUALITY AT INLET TO TRANSITION REGION
C Y.........FIN THICKNESS
C Z.........Z(9), ARRAY HOLDING PSYCROMETRIC PROPERTIES
C
C*********************************************************************
C
      PROGRAM EVAP2(TAPE10,TAPE20,TAPE30)
      REAL NF,MFA,MFR,MF2PA,MF2PR,KFIN,LENGTH,LEN2PH,LSUP,MFA2PH
      REAL MF2PAS,MFASUP,MF2ASP,MUA,KA,MFROST,NTU,MLHS,MRHS,NFSUP
      REAL MFA2PI,NFI,LE,MFA2,MFRT
      REAL Z(9)
      READ(10,*) NDTPTS,NREF,IFROST,XSET
```

```
      DO 195 NDATA=1,NDTPTS
C-----------------------------------------------------------------
C  INCOMING AIR AND REFRIGERANT PROPERTIES ARE SPECIFIED.
C-----------------------------------------------------------------
         READ(10,*) TA1,RH1,MFA,DPINIT,CPAD,HFGW
         READ(10,*) TR1,HR1,PR1,MFRT
         READ(10,*) DTIME,DTIME2,TLIMIT,PINTVL
         READ(10,*) DTSHSP,QLTYSP,INDIC
         READ(10,*) INDVEL,VSPEC
         READ(10,*) INDQL,QUAL
         READ(10,*) INDTEV,TEVAP
         READ(10,*) INDVFC,VFCE
C-----------------------------
C  COIL GEOMETRY IS SPECIFIED
C-----------------------------
         READ(10,*) ARFC,WDTH,HT,DPTH,CKTS,RWSH
         READ(10,*) DO,DI,Y,FH,KFIN,FPI
         READ(10,*) HCONT,HFOULO,HFOULI
         ROWS=CKTS
         HC=HT/RWSH
         DC=DPTH/ROWS
         LENGTH=RWSH*WDTH
         FINS=FPI*LENGTH*12
         S=1.0/(FPI*12)
C-------------------------------------------------------------------------
C VARIABLES USED IN PROPERTY SUBROUTINES ARE INITIALIZED BY CALLING
C THE FOLLOWING DATA SUBROUTINES.
C-------------------------------------------------------------------------
         CALL DATA(NREF)
         CALL DATAV(NREF)
         CALL DATAK(NREF)
         CALL DATASH(NREF)
         IF (INDTEV .EQ. 1) THEN
          TR1=TEVAP
         ELSE
          TR1=TSAT(ITR,PR1)
         END IF
         CALL SATPRP(ITR,TR1,PR1,VF,VG,HF,HFG,HG,SF,SG)
         IF (INDQL .EQ. 1) HR1=HF+QUAL*HFG
         QUAL=(HR1-HF)/HFG
C
C        EVALUATE INLET AIR PROPERTIES.
C--------------------------------------------
```

```
      Z(1)=TA1
      Z(3)=RH1
      CALL PSYC(Z,3)
      WA1=Z(4)
      HA1=Z(5)
      DPT1=Z(6)
      DENSA1=1.0/Z(9)
C
C     VARIABLES ARE INITIALIZED.
C-----------------------------------
      X2EST=0
      DTSH=0
      IWRN=0
      KK3=0
      I2=0
      IH=0
      C=0
      J1=0
      K1=0
      LK=0
      JFLAG=0
      KFLAG=0
      TCOND1=0
      TCOND2=0
      DELICE=0
      RESI2P=0
      COND=0
      LL=0
      KK=0
      AFROST=0
      XX=0
      TIME=0
      QR1HD=0
      QR2HD=0
      TWS1HD=0
      TWS2HD=0
      DPQ1HD=0
      DPQ2HD=0
      REY1HD=0
      REY2HD=0
      TCO1HD=0
      TCO2HD=0
      TCDL1=0
```

```
TCDL2=0
RI1HD=0
RI2HD=0
DEL1HD=0
DEL2HD=0
DPQRS1=0
DPQRS2=0
REYNS1=0
REYNS2=0
QRATS1=0
QRATS2=0
TWS1=0
TWS2=0
IREAD=1
DP=DPINIT
CFM=-534.03*DP+1560.7
TMFA=DENSA1*CFM*60
IF (TMFA .LT. MFA) MFA=TMFA
FINSPC=S-Y
LE=0.95
KA=-8.487E-08*TA1*TA1+3.1923E-05*TA1+1.3005E-02
PRA=1.047E-06*TA1*TA1-2.3187E-04*TA1+0.72307
MUA=-1.962E-07*TA1*TA1+8.2371E-05*TA1+3.9207E-02
CPA=CPAD+0.45*WA1
XCOMP=XSET+0.0001
MFR=MFRT/ROWS
COEFFI=0.3
PI=2.0*ATAN2(1.0,0.0)
HRX=HR1
X2=XSET
TWEST=-100
X1=(HR1-HF)/HFG
DENICE=0.9*62.5
ICTR=1
CALL FLAREA(DO,DELICE,DI,Y,FINS,HC,DC,LENGTH,ATB,AFIN,AO,
& DEQUIV,FHEQIV,AI,PI,AMF,DH)
MF2PA=MFA/AMF
REA=DH*MF2PA/MUA
ATBOUT=PI*DO*LENGTH
TW=-100
TFR=TW
TA2=TA1-2.0
WA2=WA1
```

```
      TAVI=(TA1+TA2)/2
      WAVI=(WA1+WA2)/2
C
C     THE FRACTION OF THE COIL IN THE TWO PHASE REGION IS ASSUMED.
C-----------------------------------------------------------------------
      AFCI=0.5
      AFACE=HC*LENGTH
      AMFTOT=(HC-DO)*(LENGTH-Y*FINS)
      IF (INDVEL .EQ. 1) MFA=60*DENSA1*VSPEC*AMFTOT
      IF (INDVFC .EQ. 1) MFA=60*DENSA1*VFCE*AFACE
      VFCE=MFA/(60*DENSA1*AFACE)
      SO=S-Y
      SSI=SO
      VCOILI=VFCE*S/SSI
      MF2PA=MFA/AMFTOT
      IF (TCO1HD .LE. 0.0001) GOTO 10
C
C     IF IREAD=1 A SCHEDULE OF INLET CONDITIONS ARE SPECIFIED.
C-----------------------------------------------------------------
    5 IF (IFROST .EQ. 1) THEN
       IF (IREAD .EQ. 1) READ(10,*) HR
       DELT=TIME-HR
       IF (DELT .GE. -.001) THEN
        READ(10,*) TA1,RH1,HR1,PR1,OUTLSP
        IF (INDIC .EQ. 1) DTSHSP=OUTLSP
        IF (INDIC .EQ. 2) QLTYSP=OUTLSP
        IF (TA1 .LE. 0.0001) GOTO 190
        TR1=TSAT(ITR,PR1)
        CALL SATPRP(ITR,TR1,PR1,VF,VG,HF,HFG,HG,SF,SG)
        QUAL=(HR1-HF)/HFG
        Z(1)=TA1
        Z(3)=RH1
        CALL PSYC(Z,3)
        WA1=Z(4)
        HA1=Z(5)
        DPT1=Z(6)
        DENSA1=1.0/Z(9)
        IREAD=1
       ELSE
        IREAD=0
       END IF
      END IF
   10 TA2SUP=0
```

```
      WA2SUP=0
      RH2SUP=0
      HA2SUP=0
      LENSUP=0
      QSUP=0
      TWSUP=0
      NFSUP=0
      HTASUP=0
      HTRSUP=0
      RESAS=0
      RESRS=0
      KFLAG=0
      MFASUP=0
      AMFSUP=0
      DEL2=0
      RESIC2=0
      DELICE=DEL1HD
      AFC=AFCI
      IF (AFC .LE. 0.01) AFC=1.0
      TWEST=TW
      TAV=TAVI
      WAV=WAVI
      HRX=HR1
      IF (DEL1HD .GT. FINSPC) THEN
       WRITE(20,105)
       GOTO 190
      END IF
C
C     CALCULATE FLOW AREAS.
C-----------------------------
      AFRONT=HC*LENGTH
      VFRONT=MFA/(ARFC*DENSA1*3600)
      AX=PI*(DI*DI)/4.0
      AI=PI*DI*LENGTH
      MF2PR=MFR/AX
C
C     RESET VARIABLES BETWEEN ITERATIONS.
C----------------------------------------------------------
   15 KK2=0
      X2=XSET
      X1=(HRX-HF)/HFG
      DIAOI=0
      AFCI=0
```

```
      TFRI=0
      TWI=0
      TA2I=0
      WA2I=0
      RH2I=0
      MFA2PI=0
      HA2I=0
      CONDI=0
      AMF2I=0
      HTAI=0
      NFI=0
      DELICI=0
      DELICE=DEL1HD
      HTRI=0
      RESAI=0
      RESI2P=RI1HD
      Q2PI=0
      RESRI=0
C
C     HEAT TRANSFER AVAILABLE IN THE REGION UNDER CONSIDERATION
C     IS CALCULATED.
C-----------------------------------------------------------------
   20 Q2PH=MFR*HFG*(X2-X1)*ROWS
      KK=0
      CALL FLAREA(DO,DELICE,DI,Y,FINS,HC,DC,LENGTH,ATB,AFIN,ATOT,
     & DEQUIV,FHEQIV,AI,PI,AMF,DH)
      SIGMA=AMF/AFRONT
      ASAT=4*DC*HC*SIGMA/(DO*DH*PI)
C
C     PROPERTIES OF THE AIR AT THE AVERAGE AIR TEMP. ARE CALCULATED.
C-----------------------------------------------------------------
   25 KA=-8.487E-08*TAV*TAV+3.1923E-05*TAV+1.3005E-02
      PRA=1.047E-06*TAV*TAV-2.3187E-04*TAV+0.72307
      MUA=-1.962E-07*TAV*TAV+8.2371E-05*TAV+3.9207E-02
      CPA=CPAD+0.45*WAV
      Z(1)=TAV
      Z(4)=WAV
      CALL PSYC(Z,4)
      DENSDA=1.0/Z(9)
C
      IF (INDIC .EQ. 2) THEN
       HREFSP=HF+QLTYSP*HFG
      END IF
```

```
C
C       CALCULATE SLOPE OF PROCESS LINE.
C-----------------------------------------------
        TA2EST=TA2
        TAVEST=TAV
   30   IF (JFLAG .EQ. 0) GOTO 35
        Z(1)=TFR
        Z(3)=1.0
        CALL PSYC(Z,3)
        WFR=Z(4)
        C=(WAV-Z(4))/(TAV-TFR)
        IF (C .LT. 0) C=0
C
   35   CALL HEAT2(CPA,PRA,MF2PA,MUA,DO,ASAT,HTAD,DC)
        HTAW=HTAD*HFGW*C/(CPA*LE)
        HTA=HTAD+HTAW
        CALL FINEF3(HTA,KFIN,HFGW,Y,DO,DEQUIV,FHEQIV,NF)
   40   KK=KK+1
        LEN2PH=LENGTH*AFC
        AI2PH=AI*AFC
        RESA=1.0/(HTA*AFC*(ATB+NF*AFIN)*ROWS)
        RESFM=1.0/(HTA*AFC*ATOT*ROWS)
        RESICE=RESI2P/(AO*AFC)
        RESCON=1.0/(HCONT*AFC*ATBOUT*ROWS)
        RESFLO=1.0/(HFOULO*AFC*AO*ROWS)
        RESFLI=1.0/(HFOULI*AFC*AI*ROWS)
        IF (X2 .GT. 1.0) X2=1.0
   45   CALL CLC2PH(JCR,MF2PR,DI,AI,X2,X1,HRX,PR1,TR1,HTR,RESR,
     & LENGTH,AFC,ROWS,XX,XSET,PI)
C
C       DETERMINE TOTAL RESISTANCE TO HEAT TRANSFER, AND HEAT TRANSFER.
C-------------------------------------------------------------------------
        SUMR=RESR+RESA+RESICE+RESCON+RESFLO+RESFLI
        Q=(TAV-TR1)/SUMR
        DIFFQ=Q-Q2PH
        AA=5
        HR2=HRX+Q/(MFR*ROWS)
C
C       IF KFLAG=1 THERE IS NO SUPERHEATED REGION AND EVAPORATOR
C       OUTLET IS IN REGION UNDER CONSIDERATION
C-------------------------------------------------
        IF (KFLAG .EQ. 1) THEN
         X2=(HR2-HF)/HFG
```

```
      DIFFX=X2-X1
      IF (ABS(X2-X2EST) .GT. 0.0001) THEN
       KK3=KK3+1
       X2EST=X2
       IF (KK3 .LT. 15) GOTO 45
       KK3=0
       GOTO 50
      ELSE
       KK3=0
       GOTO 50
      END IF
     END IF
     QLT=Q/(MFR*HFG*ROWS)+X1
C
C    CHECK CONVERGENCE ON HEAT TRANSFERRED IN A REGION.
C-----------------------------------------------------------------
     IF (ABS(DIFFQ) .LT. AA) THEN
      KK=0
      GOTO 50
     END IF
     IF (KK .GT. 30) THEN
      WRITE(30,*) 'CONVERGENCE NOT ATTAINED. KK=',KK
      WRITE(30,*) 'DIFFQ=',DIFFQ
      KK=0
      GOTO 50
     END IF
     IF (DIFFQ .LT. 0) THEN
       ALHS=AFC
       QLHS=Q
       J1=1
     ELSE
       ARHS=AFC
       QRHS=Q
       K1=1
     END IF
     IF ((J1 .EQ. 1) .AND. (K1 .EQ. 1)) THEN
       AFC=(Q2PH-QLHS)*((ARHS-ALHS)/(QRHS-QLHS))+ALHS
       GOTO 40
     END IF
     IF (DIFFQ .LT. 0) THEN
       AFC=ALHS+0.2*ALHS
     ELSE
       AFC=ARHS-0.2*ARHS
```

```
      END IF
      GOTO 40
C
   50 J1=0
      K1=0
      AMF2PH=AMF*AFC
      MFA2PH=MF2PA*AMF2PH
      ATOT2P=ATOT*AFC
      ATL=AFC*AO
      VFACE=MFA2PH/(DENSDA*AMF2PH*3600)
C
C     THE COIL SURFACE TEMPERATURES ARE CALCULATED.
C----------------------------------------------
      TFR=TAV-Q*RESFM
      TW=TAV-Q*(RESFM+RESICE)
      HA2=HA1-Q/MFA2PH
      LL=LL+1
C
C     IF CONVERGENCE, WITHIN SPECIFIED TOLERANCE, IS NOT ATTAINED ON
C     COIL SURFACE TEMPERATURE, THE SIMULATION IS TERMINATED.
C---------------------------------------------------------------------
      IF (LL .GT. 20) THEN
       GOTO 190
      END IF
      IF (JFLAG .NE. 0) THEN
        IF (ABS(TW-TWEST) .GT. 0.005) THEN
         TWEST=TW
         GOTO 30
        END IF
      GOTO 55
      END IF
C
C     COMPARE TWALL WITH THE DEW POINT.
C----------------------------------------
      IF (TW .LT. DPT1) THEN
        JFLAG=1
        IF (TW .LT. 32) THEN
         JFLAG=2
         HFGW=1219.0
        END IF
        TWEST=TW
        GOTO 30
      END IF
```

```
C
C       LEAVING AIR PROPERTIES ARE DETERMINED.
C-----------------------------------------------------
   55   CALL TLEAV2(KK2,HA1,HA2,C,TFR,TA1,Z,WA1,HWALL)
C
C       CONVERGENCE ON AVERAGE TEMPERATURE IS MONITORED. CONVERGENCE ON
C       AVERAGE TEMPERATURE IS THE CRIETERIA FOR COMPLETED ANALYSIS OF
C       A PARTICULAR REGION.
C-----------------------------------------------------------------------
        IF (Z(1) .LT. TR1) THEN
         Z(1)=TR1+2.0
         Z(3)=1.0
         CALL PSYC(Z,3)
        END IF
        DTLM=(TA1-Z(1))/ALOG((TA1-TR1)/(Z(1)-TR1))
        TAV=TR1+DTLM
        SAIR=(WA1-WFR)/(TA1-TFR)
        WAV=SAIR*(TAV-TFR)+WFR
        KKOUNT=KKOUNT+1
        IF (KKOUNT .GT. 30) THEN
         WRITE(30,*) 'KKOUNT REACHES ',KKOUNT
         GOTO 60
        END IF
        LL=0
        TA2=Z(1)
        IF (ABS(TAV-TAVEST) .GT. 0.005) THEN
         IF (KKOUNT .EQ. 1) TAVEST=TAV
         TAV=(0.75*TAVEST+0.25*TAV)
         WAV=SAIR*(TAV-TFR)+WFR
         GOTO 25
        END IF
   60   IF (HA2 .LT. HWALL) IWRN=1
        TA2=Z(1)
        RH2=100*Z(3)
        WA2=Z(4)
        PERC=AFC*100.0
        Z(1)=TFR
        Z(3)=1.0
        CALL PSYC(Z,3)
        WS2PH=Z(4)
C
C       SINCE THE TWO REFRIGERANT REGIONS ARE DYNAMIC A CORRECTION ON
C       THE FROST MASS IN EACH REGION BETWEEN TIME STEPS IS REQUIRED.
```

```
C-----------------------------------------------------------------------
      IF (JFLAG .NE. 2) GOTO 65
      IF ((X2 .LE. XCOMP) .AND. (AFC .LE. 1.0)) THEN
       DIST=AFC*LENGTH
       SDIFF=DIST-D1
       IF (SDIFF .GT. 0) THEN
        TCOND1=TCOND1+TCDL2*SDIFF
        TCOND2=TCOND2-TCDL2*SDIFF
       ELSE
        TCOND2=TCOND2-TCDL1*SDIFF
        TCOND1=TCOND1+TCDL1*SDIFF
       END IF
      END IF
 65   COND=0
C
C     CONDENSATION RATES AND FROST GROWTH ARE DETERMINED.
C------------------------------------------------------------
      IF (WA1 .GT. WA2) THEN
       COND=MFA2PH*(WA1-WA2)
       QLAT=COND*HFGW
      END IF
      IF (JFLAG .EQ. 2) THEN
       IF (X2 .LE. XCOMP) THEN
        DELCO1=COND*DTIME/ROWS
        TCOND1=TCOND1+DELCO1
        CALL FROST2(TCOND1,DEL1,RESIC1,Q,QLAT,S,Y,TW,INDVEL,
     & DEL1HD,VFCE,ATL,VSPEC,DENSDA,MUA,DENICE,TIME,DTIME,ROWS,
     & DPQRS1,REYNS1,QRATS1,TWS1)
       ELSE
        DELCO2=COND*DTIME/ROWS
        TCOND2=TCOND2+DELCO2
         WRITE(30,*) 'TCOND2=',TCOND2
        CALL FROST2(TCOND2,DEL2,RESIC2,Q,QLAT,S,Y,TW,INDVEL,
     & DEL2HD,VFCE,ATL,VSPEC,DENSDA,MUA,DENICE,TIME,DTIME,ROWS,
     & DPQRS2,REYNS2,QRATS2,TWS2)
       END IF
       DIAO=DO+DELICE
      END IF
      KKOUNT=0
      AFCT=AFCI+AFC
      DXC=X2-XSET
      IF (DXC .LE. 0.0001) THEN
       TAVI=TAV
```

```
        WAVI=WAV
       END IF
C
C      RESULTS OBTAINED IN THE REGION X=X1 TO XSET (IF X <XSET ) ARE
C      STORED. THE REGION X=XSET-1.0 IS THEN ANALYZED.
C-----------------------------------------------------------------------
       IF (ABS (X2-XSET) .LE. 0.0001) THEN
        IF (AFCT .GE. 1.0) THEN
         TCOND1=TCO1HD
         DPQRS1=DPQ1HD
         TWS1=TWS1HD
         REYNS1=REY1HD
         QRATS1=QR1HD
         GOTO 70
        END IF
        HRX=HF+XSET*HFG
        Q2PI=Q
        DIAOI=DIAO
        AFCI=AFC
        TFRI=TFR
        TWI=TW
        TFRI=TFR
        TA2I=TA2
        WA2I=WA2
        RH2I=RH2
        MFA2PI=MFA2PH
        HA2I=HA2
        CONDI=COND
        AMF2I=AMF2PH
        HTAI=HTA
        NFI=NF
        HTRI=HTR
        DELICE=DEL2HD
        RESAI=RESA
        RESI2P=RI2HD
        RESRI=RESR
        X2=1.0
        X1=XSET
        XX=(1.0+XSET)/2
        AFC=0.1
        Q2PH=MFR*HFG*ROWS*(X2-X1)
        GOTO 20
       END IF
```

```
      MFA2PT=MFA2PI+MFA2PH
      WA2=(WA2I*MFA2PI+WA2*MFA2PH)/MFA2PT
      HA2=(HA2I*MFA2PI+HA2*MFA2PH)/MFA2PT
      MFA2PH=MFA2PT
      IF (WA1 .GT. WA2) QLAT=MFA2PH*(WA1-WA2)*HFGW
      T2EST=TA2I
      IF (X2 .LE. XCOMP) T2EST=TA2
      CALL PROP1(HA2,WA2,T2EST,Z)
      TA2=Z(1)
      RH2=100*Z(3)
      PERC=AFCT*100.0
      DIAO=(DIAOI*AFCI+DIAO*AFC)/AFCT
      Q=Q+Q2PI
      COND=COND+CONDI
      TW=(TWI*AFCI+TW*AFC)/AFCT
      TFR=(TFRI*AFCI+TFR*AFC)/AFCT
      AMF2PH=AMF2I+AMF2PH
      HTA=(HTAI*AFCI+HTA*AFC)/AFCT
      NF=(NFI*AFCI+NF*AFC)/AFCT
      HTR=(HTRI*AFCI+HTR*AFC)/AFCT
      RESA=(RESAI*AFCI+RESA*AFC)/AFCT
      RESR=(RESRI*AFCI+RESR*AFC)/AFCT
C
      IF (KFLAG .EQ. 1) GOTO 75
   70 IF (AFCT .GE. 1.0) THEN
       IF (X2 .GE. 1.0) THEN
        TCOND2=TCO2HD
        DPQRS2=DPQ2HD
        TWS2=TWS2HD
        REYNS2=REY2HD
        QRATS2=QR2HD
       END IF
       IF (INDIC .NE. 1) THEN
        KFLAG=1
        AFC=1.0-AFCI
        X2=X1+((1.0-XSET)/(AFCT-AFCI))*AFC
        IF (X2 .LT. XSET) X2=XSET+0.01
        GOTO 20
       END IF
       DTSH=10.0*(1.0-AFCT)
       GOTO 75
      END IF
C*********************************************************************
```

```
C THE SUPERHEATED REGION IS NOW ANALYSED, USING THE E-NTU METHOD.
C THE SUPERHEATED REGION IS ASSUMED DRY.
C*************************************************************************
      DELIC2=0
      CALL FLAREA(DO,DELIC2,DI,Y,FINS,HC,DC,LENGTH,ATB,AFIN,ATOT,
     & DEQUIV,FHEQIV,AI,PI,AMF,DH)
      SIGMA=AMF/AFRONT
      ASAT=4*DC*HC*SIGMA/(DO*DH*PI)
      AMFSUP=AMF*(1-AFCT)
      MFASUP=MF2PA*AMFSUP
      LSUP=LENGTH*(1.0-AFCT)
      AISUP=AI*(1.0-AFCT)
      C=0
      TR=TR1+1
      CALL SPHT(ITR,TR,PR1,SV,CPR,GAMA,SONIC)
C
C     HEAT CAPACITIES OF THE HEAT EXCHANGING FLUIDS ARE CALCULATED.
C-------------------------------------------------------------------------
      CAIR=CPA*MFASUP
      CREF=CPR*MFR*ROWS
      IF (CREF .GT. CAIR) THEN
        CMAX=CREF
        CMIN=CAIR
      ELSE
        CMAX=CAIR
        CMIN=CREF
      END IF
      CALL HEAT2(CPA,PRA,MF2PA,MUA,DO,ASAT,HTAD,DC)
      HTAW=HTAD*HFGW*C/CPA
      HTASUP=HTAD+HTAW
      CALL FINEF3(HTASUP,KFIN,HFGW,Y,DO,DEQUIV,FHEQIV,NFSUP)
      CALL CLCSUP(TR,PR1,MF2PR,HTRSUP,RESRS,AFCT,AI,DI,ROWS)
      RESAS=1.0/(HTASUP*(1.0-AFCT)*(ATB+NFSUP*AFIN)*ROWS)
      RESCON=1.0/(HCONT*(1.0-AFCT)*ATBOUT*ROWS)
      RESASF=1.0/(HTASUP*(1.0-AFCT)*ATOT*ROWS)
      RESFLO=1.0/(HFOULO*(1.0-AFCT)*AO*ROWS)
      RESFLI=1.0/(HFOULI*(1.0-AFCT)*AI*ROWS)
      UA=1.0/(RESAS+RESRS+RESCON+RESFLO+RESFLI)
      NTU=UA/CMIN
C
C     CROSS FLOW; BOTH FLUIDS MIXED.
C-----------------------------------
      DEN1=NTU/(1.0-EXP(-NTU))
```

```
      DEN2=(CMIN/CMAX)*NTU/(1.0-EXP(-NTU*CMIN/CMAX))
      EFF=NTU/(DEN1+DEN2-1.0)
      RA=CMIN/CMAX
      QSUP=EFF*CMIN*(TA1-TR1)
      TA2SUP=TA1-EFF*(CMIN/CAIR)*(TA1-TR1)
      DTSH=EFF*(CMIN/CREF)*(TA1-TR1)
      TR2=DTSH+TR1
      WA2SUP=WA1
      Z(1)=TA2SUP
      Z(4)=WA2SUP
      CALL PSYC(Z,4)
      RH2SUP=100*Z(3)
      HA2SUP=HA1-QSUP/MFASUP
C
C     AN ESTIMATE OF AN AVG. WALL TEMP IN SUPERHEATED REGION IS MADE.
C---------------------------------------------------------------------
      TAVSUP=(TA1+TA2SUP)/2
      TWSUP=TAVSUP-QSUP*RESASF
C
   75 T2EST=AFCT*TA2+(1-AFCT)*TA2SUP
      QTOT=QSUP+Q
      Q2PH=Q
      WA2MIX=(MFA2PH*WA2+MFASUP*WA2SUP)/(MFA2PH+MFASUP)
      HA2MIX=(MFA2PH*HA2+MFASUP*HA2SUP)/(MFA2PH+MFASUP)
      MFPRT=MFA2PH+MFASUP
      CALL PROP1(HA2MIX,WA2MIX,T2EST,Z)
      TWMIX=AFCT*TW+(1.0-AFCT)*TWSUP
      TSMIX=AFCT*TFR+(1.0-AFCT)*TWSUP
      TA2MIX=Z(1)
      RH2MIX=100*Z(3)
      Z(1)=TWSUP
      Z(3)=1.0
      CALL PSYC(Z,3)
      WSSUP=Z(4)
      WSMIX=(MFA2PH*WS2PH+MFASUP*WSSUP)/MFA
      WINMIX=WA1
C-------------------------------------------------------------------
C     ITERATION ABOUT DESIRED DEGREE OF SUPERHEAT, OR OUTLET QUALITY
C     (IF SPECIFIED) IS PERFORMED AT THIS STAGE.
C-------------------------------------------------------------------
      IF (INDIC .EQ. 0) GOTO 90
      LK=LK+1
      IF (LK .GT. 20) THEN
```

```
      WRITE(30,*) 'CONV. ON OUTLET CONDITION NOT ATTAINED. LK=',LK
      GOTO 190
     END IF
     IF (INDIC .EQ. 1) THEN
       DIFF=DTSH-DTSHSP
       IF (ABS(DIFF) .LE. 0.5) THEN
        COEFFI=0.1
        J2=0
        K2=0
        GOTO 90
       END IF
     ELSE
      DIFF=HR2-HREFSP
      DTSH=HR2
      DTSHSP=HREFSP
      IF (ABS(DIFF) .LE. 0.5) THEN
       COEFFI=0.1
       J2=0
       K2=0
       GOTO 90
      END IF
     END IF
80   IF (DIFF .LT. 0) THEN
       MRHS=MFR
       DTSHR=DTSH
       J2=1
     ELSE
       MLHS=MFR
       DTSHL=DTSH
       K2=1
     END IF
     IF ((J2 .EQ. 1) .AND. (K2 .EQ. 1)) THEN
       MFR=(DTSHSP-DTSHR)*(MLHS-MRHS)/(DTSHL-DTSHR)+MRHS
       GOTO 85
     END IF
     IF (DIFF .LT. 0) THEN
       MFR=MRHS-COEFFI*MRHS
     ELSE
       MFR=MLHS+COEFFI*MLHS
     END IF
85   RESI2P=RI1HD
     DELICE=DEL1HD
     TCOND1=TCO1HD
```

```
      TCOND2=TCO2HD
      DPQRS1=DPQ1HD
      DPQRS2=DPQ2HD
      TWS1=TWS1HD
      TWS2=TWS2HD
      REYNS1=REY1HD
      REYNS2=REY2HD
      QRATS1=QR1HD
      QRATS2=QR2HD
      MF2PR=MFR/AX
      IF ((INDIC .EQ. 1) .OR. (INDIC .EQ. 2)) THEN
       JFLAG=0
       GOTO 10
      END IF
 90   CONTINUE
      MFRT=MFR*ROWS
      TCOUNT=TCOUNT+DTIME
      IF ((TIME .GT. 0.0001) .AND. (IFROST .EQ. 0))  GOTO 95
      IF ((IFROST.EQ.1).AND.((TCOUNT-PINTVL).LT.-0.0001)) GOTO 95
C
C     INPUT PARAMETERS ARE WRITTEN TO THE OUTPUT FILE.
C-----------------------------------------------------------
      WRITE(20,100) ICTR
      WRITE(20,110)
      WRITE(20,115)
      WRITE(20,160)
      WRITE(20,125)
      WRITE(20,165) ARFC,WDTH,HT,DPTH
      WRITE(20,170) FPI,RWSH,CKTS
      WRITE(20,175) HC,DC,DO,DI
      WRITE(20,180) Y,S,FH,FINS
      WRITE(20,185) ROWS,LENGTH,KFIN
      WRITE(20,120)
      WRITE(20,125)
      WRITE(20,130) TA1,RH1,PRA,KA
      WRITE(20,135) MFA,MUA,HFGW
      WRITE(20,140) TR1,HR1,QUAL,PR1
      IF (INDIC .EQ. 0) WRITE(20,145) MFRT
      IF (INDIC .EQ. 1) WRITE(20,150) DTSHSP
      IF (INDIC .EQ. 2) WRITE(20,155) QLTYSP
      IF (IWRN .EQ. 1) THEN
       WRITE(30,*) ' THE ENTHALPY OF THE LEAVING AIR IS'
       WRITE(30,*) ' LESS THAN THE SATURATED ENTHALPY CORRESPONDING'
```

```
      WRITE(30,*) ' TO THE WALL OR FROST SURFACE TEMPERTURE'
      END IF
      TEV=TR1
C
C     CALCULATE AVERAGE TEMPERATURE IN SATURATED REGION.
C-----------------------------------------------------------------
   95 DTLM=(TA1-TA2)/ALOG((TA1-TR1)/(TA2-TR1))
      TAV=TR1+DTLM
      SAIR=(WA1-WA2)/(TA1-TA2)
      WAV=SAIR*(TAV-TA2)+WA2
      Z(1)=TAV
      Z(4)=WAV
      CALL PSYC(Z,4)
      HAV=Z(5)
C
C     CALCULATE ENTHALPY AT SURFACE TEMPERATURE.
C-------------------------------------------------------------
      Z(1)=TFR
      Z(3)=1.0
      CALL PSYC(Z,3)
      HFR=Z(5)
C
C     CALCULATE ENTH. OF SATURATED AIR CORRESPONDING TO EVAP. TEMP.
C-----------------------------------------------------------------------
      Z(1)=TR1
      Z(3)=1.0
      CALL PSYC(Z,3)
      HEV=Z(5)
C
C     CALCULATE AVERAGE TEMP IN SUPERHEATED REGION.
C---------------------------------------------------------------
      TA1S=0
      IF (TA2SUP .GT. TR1) THEN
      DTLMS=(TA1-TA2SUP)/ALOG((TA1-TR1)/(TA2SUP-TR1))
      TAVSUP=TR1+DTLMS
      TA1S=TA1
      END IF
C
C     CALCULATE MIXED CONDITION AVERAGE TEMPERATURE AND ENTHALPY.
C-----------------------------------------------------------------------
      DTLMIX=(TA1-TA2MIX)/ALOG((TA1-TR1)/(TA2MIX-TR1))
      TAVMIX=TR1+DTLMIX
      WAVMIX=WA1-(TA1-TAVMIX)*(WA1-WA2MIX)/(TA1-TA2MIX)
```

```
      Z(1)=TAVMIX
      Z(4)=WAVMIX
      CALL PSYC(Z,4)
      HAVMIX=Z(5)
C
C     DETERMINE APPARATUS DEW POINT, AND THE CORRESPONDING ENTHALPY.
C-----------------------------------------------------------------------
      CALL ADPSUB(TA1,WA1,TA2,WA2,ADP,AH,ASH,TR1,JCR)
      CALL ADPSUB(TA1,WA1,TA2MIX,WA2MIX,ADPMIX,AHMIX,ASHMIX,TR1,JCR)
      CC2PH1=(ADP-TR1)/(HA1-AH)
      CC2PHA=(ADP-TR1)/(HAV-AH)
      E2PH1=(HA1-HA2)/(HA1-AH)
      E2PHA=(HAV-HA2)/(HAV-AH)
      ESUP1=(TA1S-TA2SUP)/(TA1S-ADP)
      ESUPA=(TAVSUP-TA2SUP)/(TAVSUP-ADP)
      ETEMP1=(TA1-TA2MIX)/(TA1-TR1)
      ETEMPA=(TAVMIX-TA2MIX)/(TAVMIX-TR1)
      EHOV1=(HA1-HA2MIX)/(HA1-AHMIX)
      EHOVA=(HAVMIX-HA2MIX)/(HAVMIX-AHMIX)
      CCOV1=(ADPMIX-TR1)/(HA1-AHMIX)
      CCOVA=(ADPMIX-TR1)/(HAVMIX-AHMIX)
       WRITE(30,*)  'ITR=',ITR
      IF (X2 .LE. 0.9999) THEN
       TR2=TR1
       DTSH=0
      END IF
      IF (JFLAG .NE. 2) THEN
       CALL RSLTS(TA2,WA2,RH2,TA2SUP,WA2SUP,RH2SUP,TA2MIX,WA2MIX,
     & RH2MIX,LEN2PH,LENSUP,TR2,Q2PH,QSUP,QTOT,DTSH,COND,PERC,SICE,
     & AFROST,TIME,TW,MFA,JFLAG,TWSUP,TWMIX,HA2,HA2SUP,HA2MIX,TFR,X2,
     & HTA,HTR,RESA,RESR,RESICE,HTASUP,HTRSUP,RESAS,RESRS,NF,NFSUP,
     & CC,EPS,VFCE,MFRT,QLAT)
       CALL CCEFF(CC2PH1,CC2PHA,E2PH1,E2PHA,ESUP1,ESUPA,ETEMP1,
     & ETEMPA,EHOV1,EHOVA,CCOV1,CCOVA,ADPMIX,AHMIX,HAVMIX,HAV,ADP,AH)
       GOTO 190
      END IF
C
C     IN THE CASE OF FROST ON THE COIL, THE ACCUMULATED FROST
C     AND THE NEW AIR FLOW RATE ARE DETERMINED.
C-----------------------------------------------------------------------
      MFROST=COND*DTIME
      AFROST=AFROST+MFROST
      HRX=HR1
```

```
      LK=0
      TIME=TIME+DTIME
C
C     VARIABLES CALCULATED IN THE TRANSIENT FROSTING PROCESS AT THE
C     PRESENT TIME LEVEL T ,ARE STORED FOR USE AT TIME LEVEL T+DT
C-----------------------------------------------------------------------
      TCO1HD=TCOND1
      TCO2HD=TCOND2
      RI1HD=RESIC1
      RI2HD=RESIC2
      DEL1HD=DEL1
      DEL2HD=DEL2
      DPQ1HD=DPQRS1
      DPQ2HD=DPQRS2
      TWS1HD=TWS1
      TWS2HD=TWS2
      REY1HD=REYNS1
      REY2HD=REYNS2
      QR1HD=QRATS1
      QR2HD=QRATS2
      IF (X2 .LT. XSET) AFCI=1.0
      D1=AFCI*LENGTH
      D2=(AFCT-AFCI)*LENGTH
      D3=(1.0-AFCT)*LENGTH
      TCDL1=TCOND1/D1
      IF (D2 .GT. 0) TCDL2=TCOND2/D2
      RIHOLD=(RESIC1*AFCI+RESIC2*(AFCT-AFCI))/AFCT
C
      DELICE=(DEL1*AFCI+DEL2*(AFCT-AFCI))/AFCT
      RESICE=RIHOLD/(AFCT*AO)
      DIA=DO+DELICE
      AMF2PH=(HC-DIA)*(LENGTH-(Y+2*DELICE)*FINS)*AFCT
      AMFTOT=AMF2PH+AMFSUP
      IF ((TCOUNT-PINTVL) .GE. -0.0001) THEN
        SICE=12.0*DELICE
        CALL RSLTS(TA2,WA2,RH2,TA2SUP,WA2SUP,RH2SUP,TA2MIX,WA2MIX,
     & RH2MIX,LEN2PH,LENSUP,TR2,Q2PH,QSUP,QTOT,DTSH,COND,PERC,SICE,
     & AFROST,TIME,TW,MFA,JFLAG,TWSUP,TWMIX,HA2,HA2SUP,HA2MIX,TFR,X2,
     & HTA,HTR,RESA,RESR,RESICE,HTASUP,HTRSUP,RESAS,RESRS,NF,NFSUP,
     & CC,EPS,VFCE,MFRT,QLAT)
       CALL CCEFF(CC2PH1,CC2PHA,E2PH1,E2PHA,ESUP1,ESUPA,ETEMP1,
     & ETEMPA,EHOV1,EHOVA,CCOV1,CCOVA,ADPMIX,AHMIX,HAVMIX,HAV,ADP,AH)
        TCOUNT=0
```

```
      END IF
C
C     THE FLOW IS EITHER CALCULATED FROM THE COIL PRESSURE DROP,
C     OR IT IS DETERMINED FROM A SPECIFIED FACE VELOCITY.
C------------------------------------------------------------------
      CALL PDROP(DO,DELICE,PI,Y,FINS,HC,DC,LENGTH,AFRONT,VFCE,
     & MF2PA,MUA,DENSDA,AFROST,MFA,S,DPAIR,ROWS,DIFFUS,WW,TIME,AFCT,
     & VCOILI,MFA2,DENSA1,ATOT)
      HAVG=DELICE*AFCT
      DP=ROWS*DPAIR
      DP2=DP
      IF (DP2 .LT. DPINIT) DP2=DPINIT
C
C     THE AIR FLOW RATE IS DETERMINED FROM THE FAN CHARACTERISTIC AND
C     THE PRESSURE DROP ON THE AIR SIDE.
C---------------------------------------------------------------------
      CFM=-534.03*DP2+1560.7
      MFA2=DENSA1*CFM*60
      IF (INDVEL .EQ. 1) MFA2=60*DENSA1*VSPEC*AMFTOT
      IF (INDVFC .EQ. 1) THEN
       CFM=-534.03*DP+1560.7
       MFA2=DENSA1*CFM*60
       SMFA=60*DENSA1*VFCE*AFACE
       IF (SMFA .LT. MFA2) MFA2=SMFA
      END IF
      DPCOMP=DPINIT+0.01
      BLOCKF=DELICE/(FINSPC/2)
      IF (BLOCKF .GT. 0.750) THEN
        CALL FRSTUP(COND,FINSPC,AFROST,AFCT,ATOT,ROWS,DELICE,TIME)
        GOTO 190
      ELSE
        IF (DP .LT. DPINIT) MFA=MFA2
        IF (MFA2 .LT. MFA) MFA=MFA2
      END IF
      VFCE=MFA/(60*DENSA1*AFACE)
      IF (VFCE .LT. 100) THEN
       CALL FRSTUP(COND,FINSPC,AFROST,AFCT,ATOT,ROWS,DELICE,TIME)
       GOTO 190
      END IF
      MF2PA=MFA/AMFTOT
      IF ((TIME .GE. 2.0) .AND. (I2 .EQ. 0)) THEN
       DTIME=DTIME2
       PINTVL=DTIME
```

```
        I2=1
       END IF
       DELHVG=HAVG-HAVGHD
       IF (IH .EQ. 0) THEN
       IF ((HAVG.GT.(FINSPC/4.0)).AND.(DELHVG.GT.(FINSPC/45))) THEN
        DTIME=0.5
        IH=1
       END IF
       END IF
       HAVGHD=HAVG
       IF (TIME .LT. TLIMIT) THEN
        GOTO 5
       ELSE
        CALL FRSTUP(COND,FINSPC,AFROST,AFCT,ATOT,ROWS,DELICE,
     & TIME)
       END IF
C
C      FORMATTING OF INPUT PARAMETERS IN THE OUTPUT FILE.
C-----------------------------------------------------------------
  100  FORMAT(I1)
  105  FORMAT(//,25X,'THE COIL IS FROSTED UP.',/)
  110  FORMAT(//////,25X,'I N P U T  P A R A M E T E R S')
  115  FORMAT(28X,'--------------------',/)
  120  FORMAT(15X,'ENTERING AIR AND REFRIGERANT STATES')
  125  FORMAT(13X,'----------------------------------------',/)
  130  FORMAT(10X,'TA1=',F5.2,1X,'F',3X,'RH=',F5.3,2X,'PRA=',F5.3,1X,
     & 'BTU/LB*F',3X,'KA=',F6.4,1X,'BTU/HR*F*FT',/)
  135  FORMAT(10X,'MFA=',F7.1,1X,'LB/HR',3X,'MUA=',F6.4,1X,'LB/FT*HR',
     & 3X,'HFGW=',F6.1,1X,'BTU/LB',/)
  140  FORMAT(10X,'TR=',F5.2,1X,'F',3X,'HR1=',F6.2,1X,'BTU/LBM',3X,
     & 'XIN=',F4.2,3X,'PR1=',F5.2,1X,'PSI',/)
  145  FORMAT(10X,'REFRIGERANT MASS FLOW=',F7.2,1X,'LB/HR',/)
  150  FORMAT(10X,'SPECIFIED DEGREE OF SUP.HEAT=',F6.2,' F',/)
  155  FORMAT(10X,'SPECIFIED LVG.REF.QUALITY=',E10.3,/)
  160  FORMAT(15X,'GEOMETRIC DATA. ALL DATA IN UNITS OF FEET.')
  165  FORMAT(10X,'FACE A.=',F6.3,2X,'WIDTH=',F6.3,2X,'HEIGHT=',
     & F6.3,2X,'DEPTH=',F6.3,/)
  170  FORMAT(10X,'FPI=',F4.1,2X,'ROWS H.=',F4.1,2X,'CKTS=',F4.1,/)
  175  FORMAT(10X,'ROW HEIGHT=',F6.3,2X,'ROW DEPTH=',F6.3,
     & 2X,'DIA.O.=',F6.4,2X,'DIA.I.=',F6.4,/)
  180  FORMAT(10X,'FIN THK.=',F7.5,2X,'FIN SPAC.=',F6.4,2X,'FIN H.=',
     & F6.4,2X,'FINS=',F7.1,/)
  185  FORMAT(10X,'ROWS=',F4.1,2X,'L. CKT=',F5.1,2X,'KFIN=',F6.2,
```

```
     & 1X,'BTU/HR*FT*F',////)
  190 CONTINUE
  195 CONTINUE
C
      END
C
C****************************************
C THIS SUBROUTINE CALCULATES FLOW AREAS.
C****************************************
      SUBROUTINE FLAREA(DO,DELICE,DI,Y,FINS,HC,DC,LENGTH,ATB,AFIN,ATOT,
     & DEQUIV,FHEQIV,AI,PI,AMF,DH)
      REAL LENGTH
      DIAO=DO+DELICE
      AMF=(HC-(DO+DELICE))*(LENGTH-(Y+2*DELICE)*FINS)
      ATB=PI*DIAO*(LENGTH-(Y+2*DELICE)*FINS)
      AFIN=2.0*FINS*(HC*DC-PI*(DIAO**2.0)/4.0)
      ATOT=ATB+AFIN
      DH=4*AMF*DC/ATOT
C
C THE RADIUS OF A CIRCULAR FIN OF SAME AREA AS THE FLAT PLATE FIN
C MODELLED IS CALCULATED.
C--------------------------------------------------------------------
      DEQUIV=2*(HC*DC/PI)**0.5
      FHEQIV=(DEQUIV-DO)/2.0
      RETURN
      END
C
C
C******************************************************************
C THIS SUBROUTINE CALCULATES FROST DENSITY AND FROST THERMAL
C CONDUCTIVITY AS SUGGESTED BY MALHAMMAR (1986).
C******************************************************************
      SUBROUTINE FROST2(TMCOND,DELICE,RESI,Q,QLAT,S,Y,TWALL,INDV,
     & DELHD,VFCE,ATL,VSPEC,DENSDA,MUA,DENICE,TIME,DTIME,ROWS,
     & DPQRS,REYNS,QRATS,TWS)
      REAL MUA
      IF (QLAT .LT. 0.5) RETURN
      TIME2=TIME+DTIME
      FTPREV=0
      KOUNT=0
      TWS=TWS+TWALL*DTIME
      TWAV=TWS/TIME2
      TSURF=(TWAV-32.0)/1.8
```

```
      SO=S-Y
      SS=SO-2*DELHD
      WSP=VFCE*S/(SS*60)
      IF (INDV .EQ. 1) WSP=VSPEC/60
      DPDTMP=4.325E+10*EXP(-5619/(273+TSURF))
      DPQR=DPDTMP*Q/QLAT
      DPQRS=DPQRS+DPQR*DTIME
      DPQRAV=DPQRS/TIME2
C
      DEN2=4.98E-04*(1.0+TSURF/396.0)
      QRAT=QLAT/Q
      QRATS=QRATS+QRAT*DTIME
      QRATAV=QRATS/TIME2
      TNUM=TIME2*3600*2.84E+06*QRATAV/DEN2
C
      REYNO=3600*WSP*2*SO*DENSDA/MUA
      REYNS=REYNS+REYNO*DTIME
      REYNM=REYNS/TIME2
      REYN=0.5*(REYNM+REYNO)
      IF (REYN .LT. 2600) THEN
       VM=204
       SKI=2.58E-14+1.91E-16*REYN
       GOTO 10
      END IF
      IF (REYN .LT. 22000) THEN
       VM=113.0+3.50E-02*REYN
       SKI=5.23E-13
       IF (REYN .LT. 5000) GOTO 10
    5  FT=VM*(1.05+(0.693+SKI*TNUM)**0.5)
       DENSFR=FT*DPQRAV/462.0
       IF (KOUNT .GT. 20) GOTO 15
       IF (ABS(FTPREV-FT) .LT. 0.5) GOTO 15
       KOUNT=KOUNT+1
       VM=295
       SKI=3.06E-20*FT*DENICE/DENSFR
       FTPREV=FT
       GOTO 5
      END IF
   10 FT=VM*(1.05+(0.693+SKI*TNUM)**0.5)
      DENSFR=FT*DPQRAV/462.0
   15 IF (DENSFR .GT. 380) DENSFR=380
      IF (DENSFR .LT. 90) DENSFR=90
      CONDFR=0.202*DENSFR*(1.0-DENSFR/1860.0)/(VM-0.189*DENSFR)
```

```
C
      DENSFR=0.06248*DENSFR
      CONDFR=0.64316*CONDFR
      DELICE=TMCOND/(ATL*DENSFR)
      RESI=DELICE/(CONDFR*ROWS)
      RETURN
      END
C
C
C********************************************************************
C THIS SUBROUTINE CALCULATES THE PRESSURE DROP ON THE AIR SIDE OF
C THE COIL AS A FUNCTION OF FROST ACCUMULTION ON THE COIL SURFACE.
C********************************************************************
      SUBROUTINE PDROP(DO,DELICE,PI,Y,FINS,HC,DC,LENGTH,AFRONT,VFCE,
     & MF2PA,MUA,DENSDA,AFROST,MFA,S,DPAIR,ROWS,DIFFUS,WW,TIME,AFC,
     & VCOILI,MFA2,DENSA1,ATOT)
      REAL LENGTH,MF2PA,MUA,MFA,MFA2
      HBAR=DELICE*AFC
      DEL=0
      CALL FLAREA(DO,DEL,DI,Y,FINS,HC,DC,LENGTH,ATB,AFIN,ATOT,
     & DEQUIV,FHEQIV,AI,PI,AMF,DH)
      AFACE=LENGTH*HC
      FR2=12.23*AFROST/(ATOT*ROWS)
      SO=S-Y
      SS=SO-2*HBAR
      WSP2=VFCE*S/(SS*60)
      DPAIR2=FR2*DC*DENSDA*(WSP2**2)/(2*SS*2*32.2)
      DPAIR=12*DPAIR2/62.4
      RETURN
      END
C
C********************************************************************
C     THIS SUBROUTINE ESTIMATES THE TIME REQUIRED FOR THE COIL TO
C     FROST UP COMPLETELY. WHEN THE PRESSURE DROP ACROSS THE COIL
C     BECOMES EXCESSIVE THIS SUBROUTINE IS ENTERED AND THE FINAL
C     CALCULATIONS ARE PERFORMED BEFORE TERMINATING SIMULATION.
C********************************************************************
      SUBROUTINE FRSTUP(COND,FINSPC,AFROST,AFCT,ATOT,ROWS,DELICE,
     & TIME)
      ICTR=1
      HMAX=FINSPC/2.0
      THK=12.0*HMAX
      DENSF=AFROST/(ROWS*DELICE*ATOT*AFCT)
```

```
      FROSTM=HMAX*DENSF*ATOT*AFCT*ROWS
      FRTADD=FROSTM-AFROST
      DTIMAD=FRTADD/COND
      TIME=TIME+DTIMAD
      WRITE(20,5) ICTR
      WRITE(20,10)
      WRITE(20,15)
      WRITE(20,20) FROSTM
      WRITE(20,25) TIME
      WRITE(20,30) THK
C
    5 FORMAT(I1)
   10 FORMAT(20X,'C O I L   I S   F R O S T E D   U P')
   15 FORMAT(19X,'-------------------------------------',/)
   20 FORMAT(18X,'MASS OF FROST ON COIL=',F6.2,1X,'LBM',/)
   25 FORMAT(18X,'TIME TO FROST UP COIL=',F6.2,1X,'HRS',/)
   30 FORMAT(18X,'FROST THICKNESS      =',F7.5,1X,'IN.',/)
      RETURN
      END
C
C
C*********************************************************************
C THIS SUBROUTINE FINDS THE STATE OF THE AIR LEAVING AN ELEMENT.
C IT IS USED FOR DEHUMIDIFIED AND DRY AIR.
C*********************************************************************
      SUBROUTINE TLEAV2(KK2,HA1,HA2,C,TW,TA1,Z,W1,HWALL)
      DIMENSION Z(9)
      K1=0
      K2=0
      IF (C .GT. 0) THEN
C
C     THE AIR IS DEHUMIDIFIED.
C-------------------------------
      T2NEW=(0.5*TA1+1.5*TW)/2
      Z(1)=TW
      Z(3)=1.0
      CALL PSYC(Z,3)
      HWALL=Z(5)
      IF (HA2 .LT. Z(5)) GOTO 10
C
    5 W=W1-C*(TA1-T2NEW)
      Z(1)=T2NEW
      Z(4)=W
```

```
      CALL PSYC(Z,4)
      DIFF=HA2-Z(5)
      IF (ABS(DIFF) .LT. 0.005) RETURN
      IF (DIFF .LT. 0) THEN
       K1=1
       T2RHS=T2NEW
       H2R=Z(5)
       T2NEW=T2RHS-1.0
       IF (T2NEW .LT. TW) T2NEW=TW
      ELSE
       K2=1
       T2LHS=T2NEW
       H2L=Z(5)
       T2NEW=T2LHS+1.0
       IF (T2NEW .GT. TA1) T2NEW=TA1
      END IF
      IF ((K1 .EQ. 1) .AND. (K2 .EQ. 1)) THEN
       T2NEW=T2RHS-(H2R-HA2)*(T2RHS-T2LHS)/(H2R-H2L)
      END IF
      GOTO 5
      ELSE
C
C THE AIR IS DRY
C-------------------------
      TA2=TA1-2
      Z(1)=TW
      Z(4)=W1
      CALL PSYC(Z,4)
      IF (HA2 .LT. Z(5)) GOTO 15
      CALL PROP1(HA2,W1,TA2,Z)
      END IF
      RETURN
C
   10 TOLER=0.005
      RH=1.0
      KSPEC=5
      TEV=TW
      CALL PROP2(HA2,RH,TEV,Z,KSPEC,TOLER)
      RETURN
C
   15 RH=1.0
      W=W1
      KSPEC=4
```

```
      TOLER=1.0E-05
      TEV=TA2
      CALL PROP2(W,RH,TEV,Z,KSPEC,TOLER)
      IF (Z(5) .LT. HA2) THEN
       CALL PROP1(HA2,W1,TA2,Z)
       RETURN
      ELSE
       GOTO 10
      END IF
      END
C
C
C*********************************************************************
C THIS SUBROUTINE USES ENTHALPY OR SPECIFIC HUMIDITY AT SATURATION
C TO DETERMINE THE REMAINING PROPERTIES OF AIR.
C*********************************************************************
      SUBROUTINE PROP2(W,RH,T2,Z,KSPEC,TOLER)
      DIMENSION Z(9)
      KT=0
      J1=0
      J2=0
    5 TEV=T2
      KT=KT+1
      Z(1)=TEV
      Z(3)=RH
      CALL PSYC(Z,3)
      DIFF=Z(KSPEC)-W
      IF (KT .GT. 20) RETURN
      IF (ABS(DIFF) .LT. TOLER) RETURN
      IF (DIFF .LT. 0) THEN
       T2=TEV+2
       WLHS=Z(KSPEC)
       TLHS=TEV
       J1=1
      ELSE
       T2=TEV-2
       WRHS=Z(KSPEC)
       TRHS=TEV
       J2=1
      END IF
      IF ((J1 .EQ. 1) .AND. (J2 .EQ. 1)) THEN
       T2=TLHS+(W-WLHS)*(TRHS-TLHS)/(WRHS-WLHS)
      END IF
```

```
      GOTO 5
      END
C
C*********************************************************************
C THIS SUBROUTINE USES ENTHALPY AND SPECIFIC HUMIDITY TO DETERMINE
C THE REMAINING PROPERTIES OF THE AIR.
C*********************************************************************
      SUBROUTINE PROP1(H,W,T2,Z)
      DIMENSION Z(9)
      J1=0
      J2=0
      KT=0
    5 TEV=T2
      KT=KT+1
      Z(1)=TEV
      Z(4)=W
      CALL PSYC(Z,4)
      DIFF=Z(5)-H
      IF (KT .GT. 20) RETURN
      IF (ABS(DIFF) .LT. 0.005) RETURN
      IF (DIFF .LT. 0) THEN
       T2=TEV+2
       HLHS=Z(5)
       TLHS=TEV
       J1=1
      ELSE
       T2=TEV-2
       HRHS=Z(5)
       TRHS=TEV
       J2=1
      END IF
      IF ((J1 .EQ. 1) .AND. (J2 .EQ. 1)) THEN
       T2=TLHS+(H-HLHS)*(TRHS-TLHS)/(HRHS-HLHS)
      END IF
      GOTO 5
      END
C
C*******************************************************************
C      THIS SUBROUTINE CALCULATES THE APPARATUS DEW POINT.
C*******************************************************************
      SUBROUTINE ADPSUB(TA1,WA1,TA2,WA2,ADP,AH,ASH,TE,JCR)
      DIMENSION X(9),Y(9),Z(9)
C
```

```
      X(1)=TA1
      X(4)=WA1
      Y(1)=TA2
      Y(4)=WA2
      CALL PSYC(X,4)
      CALL PSYC(Y,4)
      SHDB=(X(4)-Y(4))/(X(1)-Y(1))
      THI=201
      TLO=-201
      ITER=0
      PDSH=1000
      Z(3)=1.0
      Z(1)=Y(6)-1.0
      IF (Y(3) .LT. 0.995) GOTO 5
      Z(1)=Y(6)
      CALL PSYC(Z,3)
      GOTO 15
 5    IF (ITER .GE. 1) PDSH=DSH
      SHA=X(4)-SHDB*(X(1)-Z(1))
      CALL PSYC(Z,3)
      ITER=ITER+1
      IF (ITER .GT. 16) GOTO 15
      DSH=Z(4)-SHA
      CK=DSH/PDSH
      IF ((DSH .GT. 0) .AND. (DSH .LT. 0.00002)) GOTO 15
      IF ((DSH .GT. 0) .AND. (CK .LT. 1.0)) THI=Z(1)
      IF ((DSH .LT. 0) .OR. (CK .GT. 1)) TLO=Z(1)
      IF ((TLO .GT. -200) .AND. (THI .LT. 200)) GOTO 10
      IF ((DSH .GT. 0) .AND. (CK .LT. 1.0)) Z(1)=Z(1)-0.5
      IF ((DSH .LT. 0) .OR. (CK .GT. 1.0)) Z(1)=Z(1)+0.5
      GOTO 5
10    Z(1)=(THI+TLO)/2.0
      GOTO 5
15    ADP=Z(1)
      AH=Z(5)
      ASH=Z(4)
      IF (ADP .GT. TE) RETURN
      ADP=TE
      Z(1)=ADP
      Z(3)=1.0
      CALL PSYC(Z,3)
      AH=Z(5)
      ASH=Z(4)
```

```
      RETURN
      END
C
C*******************************************************************
C                          SUBROUTINE CALC
C THIS SUBROUTINE FINDS THE REFRIGERANT SIDE HEAT TRANSFER COEFFICIENT
C UNDER SATURATED CONDITIONS. THE CORRELATION SUGGESTED BY PIERRE
C (1955) IS USED.
C*******************************************************************
      SUBROUTINE CLC2PH(JCR,MF2PR,DI,AI,X2,X1,HR,PR,TR,HTR,RESR,
     & LENGTH,AFC,ROWS,XX,XSET,PI)
      INTEGER JCR
      REAL KRLS,MURLS,MUR,MF2PR,MURVS,KR,LENGTH
      XO=XSET
      TEVAL=TR
      MURLS=VSC(TEVAL,0)
      MURLS=3600*MURLS
      KRLS=COND(TEVAL,0)
      DX=X2-X1
C
C THE 2 PHASE REFRIGERANT HEAT TRANSFER COEFFICIENT AS SUGGESTED
C BY BO PIERRE IS CALCULATED IN SUBR. HTRDOM.
C-------------------------------------------------------------------
      CALL SATPRP(ITR,TR,PR,VF,VG,HF,HFG,HG,SF,SG)
      CALL HTRDOM(JCR,MF2PR,KRLS,MURLS,DX,DI,HFG,LENGTH,AFC,
     & HTR)
      IF (ABS(X1-XO) .LE. 0.0001) THEN
       IF (X2 .LT. XO) X2=XO
       IF (X2 .GT. 1.0) X2=1.0
       XX=(X1+X2)/2
       TR1=TR+1
       CALL CLCSUP(TR1,PR,MF2PR,HTR1PH,RES1PH,AFC,AI,DI,ROWS)
       WGT=(1.0-XX)/(1.0-XO)
       HTR=HTR*(SIN(PI*WGT/2))**2+HTR1PH*(COS(PI*WGT/2))**2
      END IF
      RESR=1.0/(HTR*AI*AFC*ROWS)
      RETURN
      END
C
C*******************************************************************
C                  SUBROUTINE CLCSUP
C THIS SUBROUTINE CALCULATES THE REFRIGERANT HEAT TRANSFER COEFFICIENT
C IN THE CASE OF SUPERHEATED REFRIGERANT.
```

```
C*************************************************************************
      SUBROUTINE CLCSUP(TR,PR,MF2PR,HTR,RESR,AFC,AI,DI,ROWS)
      REAL MF2PR,MUR,KR
      CALL VAPOR(ITR,TR,PR,VVAP,HVAP,SVAP)
      CALL SPHT(ITR,TR,PR,SV,CPR,GAMA,SONIC)
      KR=COND(TR,1.0)
      MUR=VSC(TR,1.0)
      MUR=3600*MUR
      CALL HTR1(JCR,MF2PR,CPR,KR,MUR,DI,HTR)
      RESR=1/(HTR*(1.0-AFC)*AI*ROWS)
      RETURN
      END
C
C
C*************************************************************************
C THIS SUBROUTINE CALCULATES THE 2-PHASE REFRIGERANT HEAT TRANSFER
C COEFFICIENT. PIERRE'S CORRELATION APPLICABLE FOR MASS FLOWS AND
C HEAT FLUX DENSITIES IN REGION OF INTEREST IS USED.
C*************************************************************************
      SUBROUTINE HTRDOM(JCR,MF2PR,KRLS,MURLS,DX,DI,HFGR,LENGTH,AFC,
     & HTR)
      REAL MF2PR,KRLS,MURLS,KF,LENGTH
      RE=MF2PR*DI/MURLS
      KF=778*DX*HFGR/(LENGTH*AFC)
      PARA=RE*RE*KF
      IF (PARA .GT. 0.7E+12) JCR=4
      IF (PARA .LT. 1E+09) JCR=5
      HTR=0.0009*(KRLS/DI)*PARA**0.5
      RETURN
      END
C
C
C ************************************************************************
C                         SUBROUTINE HTR1
C  THIS SUBROUTINE CALCULATES THE HEAT TRANSFER COEFFICIENT ON THE
C  REFRIGRERANT SIDE IN CASE OF SUPERHEATED REFRIGERANT. THE
C  CORRELATION IS FROM HILLER (1976).
C ************************************************************************
      SUBROUTINE HTR1(JCR,MF2PR,CPR,KR,MUR,DI,HTR)
      REAL MF2PR,MUR,KR
      AA=0.0108
      BB=0.1375
      RER=MF2PR*DI/MUR
```

```
      PR=CPR*MUR/KR
      HTR=AA*MF2PR*CPR/(PR**0.667*RER**BB)
      RETURN
      END
C
C
C***********************************************************
C THIS SUBROUTINE CALCULATES THE HEAT TRANSFER COEFFICIENT AS
C SUGGESTED BY MCQUISTON (1983).
C***********************************************************
      SUBROUTINE HEAT2(CPA,PRA,MF2PA,MUA,DO,ASAT,HTA,DC)
      REAL MF2PA,MUA
      REA=MF2PA*DO/MUA
      REB=MF2PA*DC/MUA
      RFAC=(1.0/REB)**1.2
      SJP=(1.0/REA)**0.4*(1.0/ASAT)**0.15
      SJ4=0.2675*SJP+1.325E-03
      SJ=((1.0-1280*RFAC)/(1.0-5120*RFAC))*SJ4
      HTA=MF2PA*CPA*SJ/PRA**0.667
      RETURN
      END
C
C******************************************************************
C THIS SUBROUTINE CALCULATES THE FIN EFFICIENCY OF A PLATE FIN HEAT
C EXCHANGER AS SUGGESTED BY GARDNER(1945). THE CURVE FITS ARE GIVEN
C BY TANTAKITTI(1985).
C******************************************************************
      SUBROUTINE FINEF3(HTA,KFIN,HFGW,Y,DO,DEQUIV,FHEQIV,NF)
      REAL KFIN,NF
      R=DEQUIV/DO
      X=FHEQIV*(2.0*HTA/(KFIN*Y))**0.5
      PHI1=1.03552448-0.31837607*X+0.02589744*X*X-0.00101*X**3
      PHI15=1.04188811-0.3713209*X+0.04282051*X*X-0.000637*X**3
      PHI2=1.03552448-0.45592852*X+0.08051282*X*X-0.005004*X**3
      PHI3=1.03881119-0.5195338*X+0.10275058*X*X-0.0070863*X**3
      PHI4=1.03230769-0.5540948*X+0.117016*X*X-0.0087024*X**3
      IF (R .LE. 1.0) THEN
        NF=PHI1
        GOTO 5
      END IF
      IF (R .LE. 1.5) THEN
        NF=2.0*(PHI15-PHI1)*(R-1.0)+PHI1
        GOTO 5
```

```
      END IF
      IF (R .LE. 2.0) THEN
        NF=2.0*(PHI2-PHI15)*(R-1.5)+PHI15
        GOTO 5
      END IF
      IF (R .LE. 3.0) THEN
        NF=(PHI3-PHI2)*(R-2.0)+PHI2
        GOTO 5
      END IF
      IF (R .LE. 4.0) THEN
        NF=(PHI4-PHI3)*(R-3.0)+PHI3
        GOTO 5
      END IF
      NF=PHI4
    5 CONTINUE
      RETURN
      END
C
C**********************************************************************
C THIS SUBROUTINE SENDS THE RESULTS TO THE OUTPUT FILE.
C**********************************************************************
      SUBROUTINE RSLTS(TA2,W2,RH2,TA2SUP,W2SUP,RH2SUP,TA2MIX,W2MIX,
     & RH2MIX,LEN2PH,LENSUP,TR2,Q2PH,QSUP,QTOT,DTSH,COND,PERC,SICE,
     & AFROST,TIME,TW,MFA,JFLAG,TWSUP,TWMIX,HA2,HA2SUP,HA2MIX,TFR,X2,
     & HTA,HTR,RESA,RESR,RESICE,HTASUP,HTRSUP,RESAS,RESRS,NF,NFSUP,
     & CC,EPS,VFCE,MFR,QLAT)
      REAL LEN2PH,LENSUP,NF,NFSUP,MFR
      RESICS=0
      ICTR=1
      WRITE(20,5) ICTR
      WRITE(20,10)
      WRITE(20,15)
      IF (JFLAG .EQ. 2) THEN
       WRITE(20,110)
       WRITE(20,115)
      END IF
      WRITE(20,120) TIME
      WRITE(20,130) MFA
      WRITE(20,135) VFCE
      WRITE(20,140) MFR
      WRITE(20,60)
      WRITE(20,65)
      WRITE(20,70) Q2PH,QSUP,QTOT
```

```
      WRITE(20,75) QLAT
      WRITE(20,80)
      WRITE(20,85)
      WRITE(20,90) COND
      WRITE(20,20)
      WRITE(20,25)
      WRITE(20,30) PERC
      IF (JFLAG .EQ. 2) WRITE(20,125) SICE,AFROST
      WRITE(20,35) TA2,W2,RH2
      WRITE(20,40) TFR,TW,HA2
      WRITE(20,50)
      WRITE(20,25)
      WRITE(20,35) TA2SUP,W2SUP,RH2SUP
      WRITE(20,45) TWSUP,HA2SUP
      WRITE(20,55)
      WRITE(20,25)
      WRITE(20,35) TA2MIX,W2MIX,RH2MIX
      WRITE(20,45) TWMIX,HA2MIX
      WRITE(20,95)
      WRITE(20,100)
      WRITE(20,105) TR2,DTSH,X2
      WRITE(20,145)
      WRITE(20,150)
      WRITE(20,20)
      WRITE(20,25)
      WRITE(20,155) HTA,HTR,NF
      WRITE(20,160) RESA,RESR,RESICE
      WRITE(20,50)
      WRITE(20,25)
      WRITE(20,155) HTASUP,HTRSUP,NFSUP
      WRITE(20,160) RESAS,RESRS,RESICS
C
    5 FORMAT(I1)
   10 FORMAT(///,20X,'L E A V I N G  C O N D I T I O N S.')
   15 FORMAT(18X,'========================================',/)
   20 FORMAT(18X,'S A T U R A T E D  R E G I O N.')
   25 FORMAT(16X,'----------------------------------------')
   30 FORMAT(15X,'PERCENT OF COIL IN 2-PHASE REGION=',F5.1,1X,'%',/
   35 FORMAT(15X,'TA2=',F6.2,1X,'F',3X,'W2=',E10.3,1X,'LBWV/LBDA',3X,
     & 'RH2=',F5.1,1X,'%')
   40 FORMAT(15X,'TFR=',F6.2,1X,'F',4X,'TWALL=',F6.2,1X,'F',4X,
     & 'ENTHALPY=',F5.2,' BTU/LBM',/)
   45 FORMAT(15X,'TWALL=',F6.2,1X,'F',4X,'ENTHALPY=',F5.2,' BTU/LBM',
```

```
     & /)
  50 FORMAT(18X,'S U P E R H E A T E D   R E G I O N.')
  55 FORMAT(18X,'M I X E D  C O N D I T I O N S.')
  60 FORMAT(/,20X,'H E A T   T R A N S F E R E D   (BTU/HR)')
  65 FORMAT(18X,'-------------------------------------------')
  70 FORMAT(17X,'Q2PH=',F7.1,3X,'QSUP=',F7.1,3X,'QTOT=',F7.1)
  75 FORMAT(17X,'QLAT=',F7.1,/)
  80 FORMAT(18X,'C O N D E N S A T I O N (LBWV/HR)')
  85 FORMAT(16X,'-------------------------------------')
  90 FORMAT(19X,'CONDENSATION=',F6.3,1X,'LBWV/HR',/)
  95 FORMAT(18X,'R E F R I G E R A N T   L V G.')
 100 FORMAT(16X,'--------------------------------')
 105 FORMAT(15X,'TR2=',F5.2,1X,'F',3X,'SUP.HEAT=',F5.2,1X,'F',3X,
     & 'X2=',F5.3,/)
 110 FORMAT(/,27X,'COIL IS FROSTED')
 115 FORMAT(25X,'----------------')
 120 FORMAT(/,30X,'TIME=',F5.2,1X,'HRS')
 125 FORMAT(15X,'FST.THK.=',F6.4,1X,'IN',3X,'MASS FROST=',F6.3,
     & 1X,'LBM',/)
 130 FORMAT(20X,'AIR MASS FLOW =',F8.1,1X,'LBDA/HR')
 135 FORMAT(20X,'FACE VELOCITY =',F8.1,1X,'FPM')
 140 FORMAT(20X,'REFR. FLOW     =',F7.1,1X,'LBM/HR',/)
 145 FORMAT(15X,'H E A T   T R A N S F E R   P A R A M E T E R S')
 150 FORMAT(13X,'================================================',
     & /)
 155 FORMAT(15X,'HTA=',F5.2,1X,3X,'HTR=',F6.2,3X,'FIN EFF=',F5.3,/)
 160 FORMAT(15X,'RESA=',F8.7,3X,'RESR=',F8.7,3X,'RESFST=',F8.7,/)
     RETURN
     END
C
C********************************************************************
C    THIS SUBROUTINE WRITES THE VARIOUS COIL CHARACTERISTICS AND
C    EFFECTIVENESS VALUES CALCULATED, TO THE OUTPUT FILE.
C********************************************************************
     SUBROUTINE CCEFF(CC2PH1,CC2PHA,E2PH1,E2PHA,ESUP1,ESUPA,ETEMP1,
     & ETEMPA,EHOV1,EHOVA,CCOV1,CCOVA,ADPMIX,AHMIX,HAVMIX,HAV,ADP,AH)
     WRITE(20,5)
     WRITE(20,10)
     WRITE(20,15) CC2PH1,CC2PHA,ADP
     WRITE(20,20) E2PH1,E2PHA,AH
     WRITE(20,25) ESUP1,ESUPA,HAV
     WRITE(20,30) ETEMP1,ETEMPA,ADPMIX
     WRITE(20,35) EHOV1,EHOVA,AHMIX
```

```
      WRITE(20,40) CCOV1,CCOVA,HAVMIX
C
    5 FORMAT(/,15X,'COIL CHARACTERISTICS AND COIL EFFECTIVENESS.')
   10 FORMAT(14X,'-----------------------------------------------------',/)
   15 FORMAT(15X,'CC2PH1=',F6.3,3X,'CC2PHA=',F6.3,3X,'ADP=',F5.2,1X,
     & 'F',/)
   20 FORMAT(15X,'E2PH1=',F5.3,5X,'E2PHA=',F5.3,5X,'AH=',F5.2,1X,
     & 'BTU/LBDA',/)
   25 FORMAT(15X,'ESUP1=',F5.3,5X,'ESUPA=',F5.3,5X,'HAV=',F5.2,1X,
     & 'BTU/LBDA',/)
   30 FORMAT(15X,'ETEMP1=',F5.3,4X,'ETEMPA=',F5.3,4X,'ADPMIX=',F5.2,
     & 1X,'F',/)
   35 FORMAT(15X,'EHOV1=',F5.3,5X,'EHOVA=',F5.3,5X,'AHMIX=',F5.2,1X,
     & 'BTU/LBDA',/)
   40 FORMAT(15X,'CCOV1=',F6.3,5X,'CCOVA=',F6.3,5X,'HAVMIX=',F5.2,1X,
     & 'BTU/LBDA',/)
      RETURN
      END
```

```
C*****************************************************************
C     NUMERICAL MODELLING OF DIRECT EXPANSION COILS.
C*****************************************************************
C
C     THIS PROGRAM IMPLEMENTS THE PARAMETRIC MODEL.
C----------------------------------------------------------------
C     EVAP2
      PROGRAM EVAP2
      DIMENSION Z(3,5),ECO(10),CCO(10),P(5),PO(5)
      REAL LHS,LHS1,LHS2
      CHARACTER*20 FNAME
C
C     SUBSCRIPT 1 DB  2 RH  3 SH  4 H  5 SV
C     I  1 SH + H   2 DB + RH   3 DB + SH   4 DB + H
C
      OPEN(UNIT=4,FILE='LP@D',STATUS='NEW')
      WRITE(6,5)
    5 FORMAT(4X,'INPUT FILE NAME ?       '$)
      READ(5,10) FNAME
   10 FORMAT(A10)
      OPEN(UNIT=1,FILE=FNAME,STATUS='OLD')
      WRITE(4,15) FNAME
   15 FORMAT(/,4X,'INPUT FILE NAME - ',A10,/)
C
C     EVAPORATOR INPUT
C     EAS     AIR SIDE HTE AT ECFMO
C     CC      COIL CHARACTERISTIC AT ECFMO
C
C     EVAPORATOR
C
      READ(1,*) FPI,THK,DT,TLIMIT,DC,ROWS
      WRITE(4,20) FPI,THK,DT,TLIMIT,DC,ROWS
   20 FORMAT(/,12X,'FPI',3X,'THICKNESS',2X,'TIME STEP',2X,'TIME LIMIT'
     #2X,'COIL DEPTH',2X,'ROWS',/,11X,F4.1,4X,F8.5,3X,F7.2,3X,F9.5,
     #3X,F9.2,2X,F5.1)
      SO=(1.0/FPI-12.0*THK)/12.0
      READ(1,*) NP,AFACE,AO,EASO,CCO
      WRITE(4,25) NP,AFACE,AO,EASO,CCO
   25 FORMAT(/,12X,'NP',4X,'AFACE',6X,'AO',5X,'EASO',6X,'CCO',
     #/,11X,I3,3X,F6.2,2X,F6.2,3X,F6.3,3X,F6.2)
C
      NCO=2*NP
C
```

```
C                 REFERENCE CONDITION
C     PO1=TA1, PO2=RH1, PO3=TEVAP, PO4=VFACE, PO5=XIN
C
      READ(1,*) (PO(K),K=1,NP)
      WRITE(4,30) (PO(K),K=1,NP)
   30 FORMAT(/,11X,'PO1        PO2        PO3        PO4        PO5
     #       IDNO',/,4X,5F10.3,F10.0)
      TAO=PO(1)
C
C     COEFFICIENTS IN THE EFFECTIVENESS EXPRESSION. ECO(I).
C
      READ(1,*) (ECO(K),K=1,NCO)
      WRITE(4,35) (ECO(K),K=1,NCO)
   35 FORMAT(/,12X,'ECO1        ECO2        ECO3        ECO4
     #         ECO5',/,4X,5F12.5,/,12X,'ECO6        ECO7        ECO8
     #         ECO9        ECO10',/,4X,5F12.5)
C
C     COEFFICIENTS IN THE CC EXPRESSION. CCO(I)
C
      READ(1,*) (CCO(K),K=1,NCO)
      WRITE(4,40) (CCO(K),K=1,NCO)
   40 FORMAT(/,12X,'CCO1        CCO2        CCO3        CCO4
     #         CCO5',/,4X,5F12.5,/,12X,'CCO6        CCO7        CCO8
     #         CCO9        CCO10',/,4X,5F12.5)
C
C                 OPEREATING POINT.
C     P1=TA1, P2=RH1, P3=TEVAP, P4=VFACE, P5=XIN
C
   45 READ(1,*) (P(K),K=1,NP)
      IF(P(NP).LT.0.1) STOP
      WRITE(4,50) (P(K),K=1,NP)
   50 FORMAT(/,4X,'EVAPORATOR',//,12X,'P1        P2        P3        P4
     #         P5        IDNO',/,4X,5F10.3,F10.0)
C
C     CALC ADP / COIL SURFACE TEMPERATURE
C     EAS=(H1-H2)/(H1-H3) = AIR SIDE HTE
C     CC=(DB3-TE)/(H1-H3) = COIL CHARACTERISTIC
C     REFERENCE@D 1983 EQUIPMENT, CHAP. 6
C
      TIME=0
      DPQR=0
      QRAT=0
      REYNS=0
```

```
      TWS=0
      AFROST=0^(
C
      TA=P(1)
      TE=P(3)
      VFACE=P(4)
C
      Z(1,1)=TA
      Z(1,2)=P(2)
      CALL PSYC(Z,1,2)
   55 PPHDA=60*VFACE*AFACE/Z(1,5)
      PPHDA2=PPHDA
      P(1)=TA-P(3)
      PO(1)=TAO-PO(3)
      EAS=EASO
C     WRITE(4,40) (P(K),K=1,NP)
C     WRITE(4,20) (PO(K),K=1,NP)
C     WRITE(4,25) (ECO(K),K=1,NCO)
      DO 60 I=1,NP
      EA=ECO(I)*(P(I)-PO(I))+ECO(NP+I)*(P(I)-PO(I))**2
C     WRITE(4,*) 'I=',I,' EA=',EA,' EAS=',EAS
C     WRITE(4,*) 'P(I)=',P(I),' PO(I)=',PO(I)
C     IF (EA .GT. 0.013) EA=0.013
   60 EAS=EAS+EA
C     WRITE(4,*) 'EAS=',EAS
      IF (EAS .GT. 1.00) EAS=1.00
      CC=CCO
      DO 65 I=1,NP
   65 CC=CC+CCO(I)*(P(I)-PO(I))+CCO(NP+I)*(P(I)-PO(I))**2
C
      IF (CC .LT. 0.1 ) THEN
       Z(2,1)=Z(1,1)-EAS*(Z(1,1)-TE)
       Z(2,3)=Z(1,3)
       CALL PSYC(Z,2,3)
       TIME=TLIMIT+1.0
       EASX=(Z(1,1)-Z(2,1))/(Z(1,1)-TE)
       GOTO 110
      END IF
C
      HSEN1=Z(1,1)*(.24+.444*Z(1,3))
      RHS=TE+CC*Z(1,4)
      Z(3,2)=1
      THI=1000
```

```
      TLO=-1000
      K=0
      Z(3,1)=(Z(1,1)+TE)/2
   70 K=K+1
      CALL PSYC(Z,3,2)
      IF(K.GT.32) GOTO 80
      LHS=Z(3,1)+CC*Z(3,4)
      DEL=ABS(LHS-RHS)/RHS
      IF((DEL.LT.0.00002).AND.(K.GT.2)) GOTO 80
      IF(LHS.GT.RHS) THEN
          THI=Z(3,1)
      ELSE
          TLO=Z(3,1)
      END IF
      IF((TLO.GT.-999).AND.(THI.LT.999)) GOTO 75
      ADP1=Z(3,1)
      LHS1=LHS
      IF(LHS.GT.RHS) THEN
          Z(3,1)=Z(3,1)-4
      ELSE
          Z(3,1)=Z(3,1)+4
      END IF
      IF(Z(3,1).LT.TE) Z(3,1)=TE
      GOTO 70
   75 ADP2=Z(3,1)
      LHS2=LHS
      Z(3,1)=ADP2+(RHS-LHS2)*(ADP2-ADP1)/(LHS2-LHS1)
      IF(Z(3,1).LT.TE) Z(3,1)=TE
      ADP1=ADP2
      LHS1=LHS2
      GOTO 70
C
C     CALC AIR LEAVING CONDITION
   80 HSEN3=Z(3,1)*(.24+.444*Z(3,3))
      Z(2,4)=Z(1,4)-EAS*(Z(1,4)-Z(3,4))
      H2=Z(2,4)
      SHF=(HSEN1-HSEN3)/(Z(1,4)-Z(3,4))
      IF(ABS(SHF).GT.0.001) THEN
        Z(2,3)=Z(1,3)-(Z(1,3)-Z(3,3))*(Z(1,4)-Z(2,4))/(Z(1,4)-Z(3,4)
      ELSE
        Z(2,3)=Z(1,3)
      END IF
      CALL PSYC(Z,2,1)
```

```
      HSEN2=Z(2,1)*(.24+.444*Z(2,3))
      LHS=(HSEN1-HSEN2)/(Z(1,4)-Z(2,4))
      DEL=ABS(LHS-SHF)/SHF
      IF(DEL.LT.0.00002) GOTO 105
   85 IF(Z(2,2).LT.0.999) GOTO 105
      EASX=(Z(1,4)-Z(2,4))/(Z(1,4)-Z(3,4))
      CCX=(Z(3,1)-TE)/(Z(1,4)-Z(3,4))
C     ALTERNATE METHOD
C
      WRITE(4,120) K,EAS,EASX,CC,CCX
      WRITE(4,125)
      DO 90 I=2,4
   90 WRITE(4,135) I,(Z(I,J),J=1,5)
C
      Z(2,2)=.999
      THI=1000
      TLO=-1000
      K=0
   95 CALL PSYC(Z,2,2)
      DEL=ABS(Z(2,4)-H2)/H2
      K=K+1
      IF((DEL.LT.0.0002).OR.(K.GT.16)) GOTO 105
      IF(Z(2,4).LT.H3) THEN
          TLO=Z(2,1)
      ELSE
          THI=Z(2,1)
      END IF
      IF((TLO.GT.-999).AND.(THI.LT.999)) GOTO 100
      DB21=Z(2,1)
      H21=Z(2,4)
      IF(Z(2,4).LT.H2) THEN
          Z(2,1)=Z(2,1)+1
      ELSE
          Z(2,1)=Z(2,1)-1
      END IF
      GOTO 95
  100 DB22=Z(2,1)
      H22=Z(2,4)
      Z(2,1)=DB22+(H2-H22)*(DB22-DB21)/(H22-H21)
      DB21=DB22
      H21=H22
      GOTO 95
  105 EASX=(Z(1,4)-Z(2,4))/(Z(1,4)-Z(3,4))
```

```
      CCX=(Z(3,1)-TE)/(Z(1,4)-Z(3,4))
  110 QTOT=(Z(1,4)-Z(2,4))*PPHDA
      H2O=(Z(1,3)-Z(2,3))*PPHDA
      VISC=-1.962E-07*Z(1,1)*Z(1,1)+8.2371E-05*Z(1,1)+3.9207E-02
      QLAT=H2O*1219.0
      TIME=TIME+DT
      CCX=(Z(3,1)-TE)/(Z(1,4)-Z(3,4))
      IF ((H2O .GT. 0) .AND. (Z(3,1) .LT. 32.0)) THEN
      CALL FRST(QTOT,QLAT,Z(3,1),H2O,DPQR,QRAT,REYNS,AFROST,AFACE,
     #VFACE,PPHDA,TIME,DT,SO,THK,DC,AO,HICE,Z(1,5),VISC,ROWS,BLF,TWS)
      ELSE
      TIME=TLIMIT+1.0
      END IF
      IF (BLF .GT. 0.75) THEN
       WRITE(4,140)
       CALL FRSTUP(H2O,SO,AFROST,AO,ROWS,HICE,TIME)
       GOTO 45
      END IF
      HPRT=HICE*12.0
CCC   IF(H2O.LT.0) H2O=0
      WRITE(4,115) TIME,HPRT,AFROST,PPHDA2
  115 FORMAT(/,8X,'TIME',4X,'FROST THK',4X,'ACC.FROST',5X,'PPHDA',/,
     #7X,F6.2,4X,F6.4,6X,F6.2,7X,F7.1)
      WRITE(4,120) K,EAS,EASX,CC,CCX,QTOT,H2O
  120 FORMAT(/,8X,'K        EAS       EASX       CC        CCX       QTOT
     #        H2O',/,4X,I5,4F10.4,F10.0,F10.3)
      WRITE(4,125)
  125 FORMAT(/,8X,'I',9X,'T         RH         SH         H         SV')
      DO 130 I=1,3
  130 WRITE(4,135) I,(Z(I,J),J=1,5)
  135 FORMAT(/,4X,I5,F10.1,F10.3,F10.5,F10.2,F10.4)
      WRITE(4,140)
  140 FORMAT(///)
C 115 FORMAT(1H1)
      VFACE2=VFACE
C
C     THE CC PARAMETER IS CORRECTED FOR ANY CHANGE IN THE AIR FLOW
C
      P(4)=VFACE
C
      IF (TIME .LT. TLIMIT) GOTO 55
      GOTO 45
      END
```

```
      SUBROUTINE FRST(QTOT,QLAT,ADP,H2O,DPQR,QRAT,REYNS,AFROST,AFACE,
     #VFACE,MFA,TIME,DT,SO,Y,DC,AO,H,SV1,VISC,ROWS,BLOCKF,TWS)
      REAL MFA
      DENSA1=1.0/SV1
      AFROST=AFROST+H2O*DT
      TWS=TWS+ADP*DT
      TS=(TWS/TIME-32.0)/1.8
      SS=SO-2*H
      WSP=VFACE*(SO+Y)/(SS*60)
      DPDTMP=4.325E+10*EXP(-5619/(273+TS))
      DPQR=DPQR+(DPDTMP*QTOT/QLAT)*DT
      DPQRAV=DPQR/TIME
      DEN2=4.98E-04*(1.0+TS/396)
      QRAT=DT*QLAT/QTOT+QRAT
      TNUM=3600*2.84E+06*QRAT/DEN2
      REYNO=3600*WSP*2*SO*DENSA1/VISC
      REYNS=REYNS+DT*REYNO
      REYN=0.5*(REYNS/TIME+REYNO)
      IF (REYN .LT. 2600) THEN
       VM=204
       SKI=2.58E-14+1.91E-16*REYN
      ELSE
       VM=113.0+3.50E-02*REYN
       SKI=5.23E-13
      END IF
      FT=VM*(1.05+(0.693+SKI*TNUM)**0.5)
      DENSFR=FT*DPQRAV/462.0
      IF (DENSFR .GT. 380) DENSFR=380
      IF (DENSFR .LT. 90) DENSFR=90
      DENSFR=0.06248*DENSFR
      H=AFROST/(AO*ROWS*DENSFR)
      FR=12.23*AFROST/(AO*ROWS)
      SS=SO-2*H
      WSP=VFACE*(SO+Y)/(SS*60)
      DPAIR=FR*DC*DENSA1*WSP*WSP/(2*SS*2*32.2)
      DPAIR=(12*DPAIR/62.4)*ROWS
      WRITE(4,5) DENSFR,DPAIR
    5 FORMAT(/,12X,'FROST DENSITY',4X,'PRES.DROP',/,10X,F11.3,8X,F8.3)
      CFM=-534.03*DPAIR+1560.7
      TMFA=[illegible]1*CFM*60
      SMFA=60*DENSA1*VFACE*AFACE
      IF (SMFA .LT. TMFA) TMFA=SMFA
```

```
      IF (TMFA .LT. MFA) MFA=TMFA
      BLOCKF=H/(SO/2)
      VFACE=MFA/(60*DENSA1*AFACE)
      RETURN
      END

      SUBROUTINE FRSTUP(H20,SO,AFROST,AO,ROWS,HICE,TIME)
      HMAX=SO/2
      HINCH=12.0*HMAX
      DENSF=AFROST/(ROWS*HICE*AO)
      FROSTM=HMAX*DENSF*AO*ROWS
      FRTADD=FROSTM-AFROST
      DTIMAD=FRTADD/H20
      TIME=TIME+DTIMAD
      WRITE(4,5)
      WRITE(4,10) FROSTM
      WRITE(4,15) TIME
      WRITE(4,20) HINCH
    5 FORMAT(18X,'THE COIL IS BLOCKED',//)
   10 FORMAT(15X,'MASS OF FROST ON COIL=',F6.2,1X,'LBM',/)
   15 FORMAT(15X,'TIME TO FROST UP COIL=',F6.2,1X,'HRS',/)
   20 FORMAT(15X,'FROST THICKNESS       =',F7.5,1X,'IN.',/)
      RETURN
      END

      SUBROUTINE PSYC(Z,N,I)
C     SUBSCRIPT 1 DB  2 RH  3 SH  4 H  5  SV
C     I  1 SH + H   2 DB + RH   3 DB + SH   4 DB + H
      DIMENSION Z(3,5)
C     PMIX=1.
C     PSF=2116.224
C     RA=53.352
C     TAZ=459.67
      IF(I .NE. 4) GO TO 10
      IF(Z(N,3) .GT. 0.001) GO TO 10
      WRITE(4,5) Z(N,1),Z(N,3)
    5 FORMAT(5X,'SH .LT. 0.001',5X,2(F10.5,5X))
      Z(N,3)=0.001
   10 GO TO (15,35,20,30) I
   15 Z(N,1)=(Z(N,4)-1061*Z(N,3))/(.24+.444*Z(N,3))
      GOTO 25
   20 PW=Z(N,3)/(.62198+Z(N,3))
```

```
25 PWS=PVAP(Z(N,1))
   SHS=.62198*PWS/(1-PWS)
   IF(Z(N,3).GT.SHS) Z(N,3)=SHS
   PW=Z(N,3)/(.62198+Z(N,3))
   Z(N,2)=PW/PWS
   GO TO 40
30 Z(N,3)=(Z(N,4)-.24*Z(N,1))/(1061.+.444*Z(N,1))
   GO TO 25
35 PWS=PVAP(Z(N,1))
   PW=Z(N,2)*PWS
   Z(N,3)=.62198*PW/(1-PW)
   SHS=.62198*PWS/(1-PWS)
40 SHF=Z(N,3)/SHS
   Z(N,5)=53.352*(Z(N,1)+459.67)*(1.+1.6078*Z(N,3))/2116.224
   Z(N,4)=.24*Z(N,1)+Z(N,3)*(1061.+.444*Z(N,1))
45 RETURN
   END
C    H SENS =Z(N,1)*(.24+.444*Z(N,3))
C    H LAT =Z(N,3)*1061.


   FUNCTION PVAP(TDB)
   TAZ=459.688
   TFREZ=491.688
   TA=TDB+TAZ
   TR=TFREZ/TA
   RTR=1./TR
   IF(TDB.LT.32.) GO TO 5
   ALOGP=10.79586*(1.-TR) + 5.02808*ALOG10(TR) + 1.50474E-4*
  2(1.-10.**(8.29692*(1.-RTR))) + 0.42873E-3*(10.**(4.76955*(1.-TR)
  3-1.) - 2.219583
   GO TO 10
 5 ALOGP=-9.096936*(TR-1.) - 3.56654*ALOG10(TR) + .876817*(1.-RTR)
  2- 2.2195983
10 PVAP=10.**ALOGP
   RETURN
   END
```

```
C
C     PROPERTIES OF R-12 AND R22
      SUBROUTINE DATA(NR)
      COMMON/QRS/I,
     1       AVP,BVP,CVP,DVP,EVP,FVP,
     2       ACV,BCV,CCV,DCV,ECV,FCV,
     3       AL,BL,CL,DL,EL,
     4       R,B,A2,B2,C2,A3,B3,C3,A4,B4,
     5       C4,A5,B5,C5,A6,B6,C6,K,ALPHA,
     6       X,Y,CPR,TFR,PC,TC,VC,AA,BB
      REAL K
      IF(NR.EQ.12) I=1
      IF(NR.EQ.22) I=2
      GOTO(5,10),I
    5 AL=34.84
      BL=0.02696
      CL=0.834921
      DL=6.02683
      EL=-0.655549E-5
      AVP=39.88381727
      BVP=-3436.632228
      CVP=-12.47152228
      DVP=0.004730442442
      EVP=0.0
      FVP=0.0
      R=0.088734
      B=0.0065093886
      A2=-3.409727134
      B2=0.00159434848
      C2=-56.7627671
      A3=0.06023944654
      B3=-1.879618431E-5
      C3=1.311399484
      A4=-0.000548737007
      B4=0.0
      C4=0.0
      A5=0.0
      B5=3.468834E-9
      C5=-2.54390678[illegible]
      A6=0.0
      B6=0.0
      C6=0.0
```

```
      K=5.475
      ALPHA=0.0
      ACV=0.0080945
      BCV=0.000332662
      CCV=-2.413896E-7
      DCV=6.72363E-11
      ECV=0.0
      FCV=0.0
      X= 39.55655122
      Y=-0.0165379361
      PC=596.9
      TC=693.3
      VC=0.02870
      CPR=0.0
      TFR=459.7
      AA=120.
      BB=312.
      RETURN
   10 AL=32.76
      BL=54.6344093
      CL=36.74892
      DL=-22.2925657
      EL=20.47328862
      AVP=29.35754453
      BVP=-3845.193152
      CVP=-7.86103122
      DVP=0.002190939044
      EVP=0.445746703
      FVP=686.1
      R=0.124098
      B=0.002
      A2=-4.353547
      B2=0.002407252
      C2=-44.066868
      A3=-0.017464
      B3=7.62789E-5
      C3=1.483763
      A4=0.002310142
      B4=-3.605723E-6
      C4=0.0
      A5=-3.724044E-5
      B5=5.355465E-8
      C5=-1.845051E-4
```

```
      A6=1.363387E+8
      B6=-1.672612E+5
      K=4.2
      ALPHA=548.2
      CPR=0.0
      TFR=459.67
      ACV=0.02812836
      BCV=2.255408E-4
      CCV=-6.509607E-8
      DCV=0.0
      ECV=0.0
      FCV=257.341
      X=62.4009
      Y=-0.0453335
      PC=721.906
      TC=664.5
      VC=0.030525
      AA=120.
      BB=388.
      RETURN
      END

      FUNCTION TSAT(ITR,PSAT)
      COMMON/QRS/I,
     1        AVP,BVP,CVP,DVP,EVP,FVP,
     2        ACV,BCV,CCV,DCV,ECV,FCV,
     3        AL,BL,CL,DL,EL,
     4        R,B,A2,B2,C2,A3,B3,C3,A4,B4,
     5        C4,A5,B5,C5,A6,B6,C6,K,ALPHA,
     6        X,Y,CPR,TFR,PC,TC,VC,AA,BB
      REAL LE10,K
      DATA LE10/2.302585093/
      IF(PSAT.LE.0.0) GOTO 15
      PLOG=ALOG10(PSAT)
      TR=AA*PLOG+BB
      ITER=0
    5 TRO=TR
      ITER=ITER+1
      IF(ITER.GT.10) GOTO 10
      C=ALOG10(ABS(FVP-TRO))
      F=AVP+BVP/TRO+CVP*ALOG10(TRO)+DVP*TRO+EVP*((FVP-
     1TRO)/TRO)*C-PLOG
```

```
      FP=-BVP/TRO**2+CVP/(LE10*TRO)+DVP-EVP*(1./(LE10*TRO)+
     1FVP*C/TRO**2)
      TR=TRO-F/FP
      IF(ABS(TR-TRO).GT..01) GOTO 5
      TSAT=TR-TFR
      RETURN
   10 TSAT=TR-TFR
      WRITE(4,20) ITR,ITER,PSAT
      RETURN
   15 TSAT=0
      WRITE(4,20) ITR,ITER,PSAT
   20 FORMAT(/,10X,'ERROR - TSAT'
     2,/,12X,'ITR ITER           PSAT',/,10X,2I5,F14.3,/)
      RETURN
      END


      FUNCTION SPVOL(ITR,TF,PPSIA)
      COMMON/QRS/I,
     1         AVP,BVP,CVP,DVP,EVP,FVP,
     2         ACV,BCV,CCV,DCV,ECV,FCV,
     3         AL,BL,CL,DL,EL,
     4         R,B,A2,B2,C2,A3,B3,C3,A4,B4,
     5         C4,A5,B5,C5,A6,B6,C6,K,ALPHA,
     6         X,Y,CPR,TFR,PC,TC,VC,AA,BB
      REAL K
      T=TF+TFR
      IF(T.LE.0.0) GOTO 15
      TFSAT=TSAT(ITR,PPSIA)
      IF(TF.LT.(TFSAT-0.002)) GOTO 15
      IF(PPSIA.LE.0.0) GOTO 15
      ARG=-K*T/TC
      IF (ARG .LT. -100) ARG=-100
      ES0=EXP(ARG)
      ES1=PPSIA
      ES2=R*T
      ES3=A2+B2*T+C2*ES0
      ES4=A3+B3*T+C3*ES0
      ES5=A4+B4*T+C4*ES0
      ES6=A5+B5*T+C5*ES0
      ES7=A6+B6*T+C6*ES0
      ES32=2.*ES3
      ES43=3.*ES4
```

```
      ES54=4.*ES5
      ES65=5.*ES6
      VN=R*T/PPSIA
      ITER=0
    5 ITER=ITER+1
      IF(ITER.GT.30) GOTO 10
      V=VN
      V2=V**2
      V3=V**3
      V4=V**4
      V5=V**5
      V6=V**6
      ARG=-ALPHA*(V+B)
      IF  (ARG.LT.-100) THEN
        ITR=ARG
        ARG=-100
      END IF
      EMAV=EXP(ARG)
      F=ES1-ES2/V-ES3/V2-ES4/V3-ES5/V4-ES6/V5-ES7*EMAV
      FV=ES2/V2+ES32/V3+ES43/V4+ES54/V5+ES65/V6+ES7*ALPHA*EMAV
      VN=V-F/FV
      IF(ABS((VN-V)/V).GT.1.E-06) GOTO 5
      SPVOL=VN+B
      RETURN
   10 SPVOL=VN+B
      WRITE(4,20) ITR,ITER,T,TF,PPSIA
      RETURN
   15 SPVOL=0
      WRITE(4,20) ITR,ITER,T,TF,PPSIA
   20 FORMAT(/,10X,'ERROR - SPVOL'
     2,/,12X,'ITR ITER                T               TF'
     3,9X,'PPSIA',/,10X,2I5,3F14.3,/)
      RETURN
      END

      SUBROUTINE SATPRP(ITR,TF,PSAT,VF,VG,HF,HFG,HG,SF,SG)
      COMMON/QRS/I,
     1        AVP,BVP,CVP,DVP,EVP,FVP,
     2        ACV,BCV,CCV,DCV,ECV,FCV,
     3        AL,BL,CL,DL,EL,
     4        R,B,A2,B2,C2,A3,B3,C3,A4,B4,
     5        C4,A5,B5,C5,A6,B6,C6,K,ALPHA,
```

```
6           X,Y,CPR,TFR,PC,TC,VC,AA,BB
      REAL J,KTDTC,LE10,K,L10E
      DATA J,LE10/0.185053,2.302585093/
      L10E=0.4342944819
      T=TF+TFR
      IF(T.LE.0.0) GOTO 65
      IF(T.GE.TC) GOTO 65
      DEL = TC -T
      GOTO(5,10),I
    5 PSAT=10.**(AVP+BVP/T+CVP*ALOG10(T)+DVP*T)
      GOTO 15
   10 PSAT=10.**(AVP+BVP/T+CVP*ALOG10(T)+DVP*T+EVP*
     1((FVP-T)/T)*ALOG10(FVP-T))
   15 VG=SPVOL(ITR,TF,PSAT)
      GOTO(20,25),I
   20 TCMT=TC-T
      VF=1./(AL+BL*TCMT+CL*TCMT**(1./2.)+DL*TCMT**(1./3.)+
     1EL*TCMT**2)
      GOTO 30
   25 TR1=1.-T/TC
      VF=1./(AL+BL*TR1**(1./3.)+CL*TR1**(2./3.)+DL*TR1+
     1EL*TR1**(4./3.))
   30 GOTO(35,40),I
   35 HFG=(VG-VF)*PSAT*LE10*(-BVP/T+CVP/LE10+DVP*T)*J
      GOTO 45
   40 HFG=(VG-VF)*PSAT*LE10*(-BVP/T+CVP/LE10+DVP*T-
     1EVP*(L10E+FVP*ALOG10(FVP-T)/T))*J
   45 SFG=HFG/T
      T2=T**2
      T3=T**3
      T4=T**4
      VR=VG-B
      VR2=2.*VR**2
      VR3=3.*VR**3
      VR4=4.*VR**4
      KTDTC=K*T/TC
      ARG=-KTDTC
      IF (ARG .LT. -100) ARG=-100
      EKTDTC=EXP(ARG)
      ARG=-ALPHA*VG
      IF (ARG .LT. -100) ARG=-100
      EMAV=EXP(ARG)
      H1=ACV*T+BCV*T2/2.+CCV*T3/3.+DCV*T4/4.-FCV/T
```

```
      H2=J*PSAT*VG
      H3=A2/VR+A3/VR2+A4/VR3+A5/VR4
      H4=C2/VR+C3/VR2+C4/VR3+C5/VR4
      S1=ACV*ALOG(T)+BCV*T+CCV*T2/2.+DCV*T3/3.-FCV/(2.*T2)
      S2=J*R*ALOG(VR)
      S3=B2/VR+B3/VR2+B4/VR3+B5/VR4
      S4=H4
      GOTO(55,50),I
   50 H3=H3+A6/ALPHA*EMAV
      S3=S3+B6/ALPHA*EMAV
   55 HG=H1+H2+J*H3+J*EKTDTC*(1.+KTDTC)*H4+X
      SG=S1+S2-J*S3+J*EKTDTC*K/TC*S4+Y
      HF=HG-HFG
      SF=SG-SFG
      IF (DEL.GT.10.) RETURN
      WRITE(4,60) ITR,DEL,VF,VG,HF,HG
   60 FORMAT(10X,'CAUTION TCRIT-T < 10F',/,17X,'ITR',5X,'DEL',8X,
     1'VF',8X,'VG',8X,'HF',8X,'HG',/,10X,I5,5F10.5)
      RETURN
   65 WRITE(4,70) ITR,T,TF
   70 FORMAT(/,10X,'ERROR - SATPRP'
     2,/,12X,'ITR              T              TF'
     3,/,10X,I5,2F14.3,/)
      RETURN
      END


      SUBROUTINE VAPOR(ITR,TF,PPSIA,VVAP,HVAP,SVAP)
      COMMON/QRS/I,
     1          AVP,BVP,CVP,DVP,EVP,FVP,
     2          ACV,BCV,CCV,DCV,ECV,FCV,
     3          AL,BL,CL,DL,EL,
     4          R,B,A2,B2,C2,A3,B3,C3,A4,B4,
     5          C4,A5,B5,C5,A6,B6,C6,K,ALPHA,
     6          X,Y,CPR,TFR,PC,TC,VC,AA,BB
      REAL J,KTDTC,LE10,K
      J=0.185053
      T=TF+TFR
      IF(T.LE.0.0) GOTO 15
      TFSAT=TSAT(ITR,PPSIA)
      IF(TF.LT.(TFSAT-0.002)) GOTO 15
      IF(PPSIA.LE.0.0) GOTO 15
      VVAP=SPVOL(ITR,TF,PPSIA)
```

```
      T2=T**2
      T3=T**3
      T4=T**4
      VR=VVAP-B
      VR2=2.*VR**2
      VR3=3.*VR**3
      VR4=4.*VR**4
      KTDTC=K*T/TC
      ARG=-KTDTC
      IF (ARG .LT. -100) ARG=-100
      EKTDTC=EXP(ARG)
      ARG=-ALPHA*VVAP
      IF (ARG .LT. -100) ARG=-100
      EMAV=EXP(ARG)
      H1=ACV*T+BCV*T2/2.+CCV*T3/3.+DCV*T4/4.-FCV/T
      H2=J*PPSIA*VVAP
      H3=A2/VR+A3/VR2+A4/VR3+A5/VR4
      H4=C2/VR+C3/VR2+C4/VR3+C5/VR4
      S1=ACV*ALOG(T)+BCV*T+CCV*T2/2.+DCV*T3/3.-FCV/(2.*T2)
      S2=J*R*ALOG(VR)
      S3=B2/VR+B3/VR2+B4/VR3+B5/VR4
      S4=H4
      GOTO(10,5),I
    5 H3=H3+A6/ALPHA*EMAV
      S3=S3+B6/ALPHA*EMAV
   10 HVAP=H1+H2+J*H3+J*EKTDTC*(1.+KTDTC)*H4+X
      SVAP=S1+S2-J*S3+J*EKTDTC*K/TC*S4+Y
      RETURN
   15 WRITE(4,20) ITR,T,TF,PPSIA,TFSAT
   20 FORMAT(/,10X,'ERROR - VAPOR'
     2,/,12X,'ITR              T               TF'
     3,9X,'PPSIA              TFSAT',/,10X,I5,5F14.3,/)
      RETURN
      END

      SUBROUTINE SPHT(ITR,TF,PPSIA,CV,CP,GAMA,SONIC)
      COMMON/QRS/I,
     1         AVP,BVP,CVP,DVP,EVP,FVP,
     2         ACV,BCV,CCV,DCV,ECV,FCV,
     3         AL,BL,CL,DL,EL,
     4         R,B,A2,B2,C2,A3,B3,C3,A4,B4,
     5         C4,A5,B5,C5,A6,B6,C6,K,ALPHA,
```

```
     6          X,Y,CPR,TFR,PC,TC,VC,AA,BB
      REAL K
      T=TF+TFR
      IF(T.LE.0.0) GOTO 20
      TFSAT=TSAT(ITR,PPSIA)
      IF(TF.LT.(TFSAT-0.002)) GOTO 20
      IF(PPSIA.LE.0.0) GOTO 20
      VVAP=SPVOL(ITR,TF,PPSIA)
      V1=VVAP-B
      V2=V1*V1
      V3=V2*V1
      V4=V3*V1
      V5=V4*V1
      V6=V5*V1
      ARG=-K*T/TC
      IF (ARG .LT. -100) ARG=-100
      EKTTC=EXP(ARG)
      ARG=-ALPHA*VVAP
      IF (ARG .LT. -100) ARG=-100
      GOTO(5,10),I
    5 FDPDV=0
      FDPDT=0
      GOTO 15
   10 FDPDV=-ALPHA*EXP(ARG)*(A6+B6*T)
      FDPDT=B6*EXP(ARG)
   15 DPDV=-R*T/V2-2.*(A2+B2*T+C2*EKTTC)/V3-3.*(A3+B3*
     1     T+C3*EKTTC)/V4-4.*(A4+B4*T+C4*EKTTC)/V5-5.*(A5
     2     +B5*T+C5*EKTTC)/V6+FDPDV
      DPDT=R/V1+(B2-K*C2*EKTTC/TC)/V2+(B3-K*C3*
     1     EKTTC/TC)/V3+(B4-K*C4*EKTTC/TC)/V4+(B5-K
     2     *C5*EKTTC/TC)/V5+FDPDT
      FCCV=0
      CV=ACV+BCV*T+CCV*T**2+DCV*T**3+FCV/T**2-(0.185053*
     1   K**2*T*EKTTC/TC**2)*(C2/V1+C3/(2.*V2)+C4/(3.*V3)
     2   +C5/(4.*V4)+FCCV)
      CP=CV-0.185053*T*DPDT**2/DPDV
      GAMA=CP/CV
      SONIC=VVAP*SQRT(857.36091*T*DPDT**2/CV-4633.056*DPDV)
      RETURN
   20 WRITE(4,25) ITR,T,TF,PPSIA,TFSAT
   25 FORMAT(/,10X,'ERROR - SPHT'
     2,/,12X,'ITR                T              TF'
     3,9X,'PPSIA          TFSAT',/,10X,I5,5F14.3,/)
```

```
      RETURN
      END

      FUNCTION VSC(T,X)
C     VISCOSITY  UNITS = LBM/(FT.SEC)   CONVERTED FROM LBM/(FT.HR)
      COMMON/AV/CWF(14),CWG(14)
      COMMON/BV/TW(12)
      VSCF=WV(CWF,T)
      IF(X.GT.0.0001) GOTO 5
      VSC=VSCF/3600.
      RETURN
    5 VSCG=WV(CWG,T)
      ITR=999
      CALL SATPRP(ITR,T,PSAT,VF,VG,HF,HFG,HG,SF,SG)
      VSC=(1.-X)*VF*VSCF+X*VG*VSCG
      VM=VF+X*(VG-VF)
      VSC=VSC/VM
      VSC=VSC/3600.
      RETURN
      END

      FUNCTION WV(C,T)
C     EVALUATE FUNCTION USING COEFFICIENTS DETERMINED BY
C     SPLINE FIT INTERPOLATION
      DIMENSION C(14)
      COMMON/BV/TW(12)
      WV=0.
      DO 5 I=1,11
    5 WV=WV+USF(T,TW(I))*C(I)*(T-TW(I))**3
      WV=WV+C(12)*T**2+C(13)*T+C(14)
      RETURN
      END

      FUNCTION COND(T,X)
C     CONDUCTIVITY UNITS = LBM/(FT.SEC)   CONVERTED FROM LBM/(FT.HR)
      COMMON/AK/CWF(14),CWG(14)
      COMMON/BK/TW(12)
      CONDF=WK(CWF,T)
      IF(X.GT.0.0001) GOTO 5
      COND=CONDF
```

```
      RETURN
    5 CONDG=WK(CWG,T)
      COND=(1.-X)*CONDF+X*CONDG
      RETURN
      END

      FUNCTION WK(C,T)
C     EVALUATE FUNCTION USING COEFFICIENTS DETERMINED BY
C     SPLINE FIT INTERPOLATION
      DIMENSION C(14)
      COMMON/BK/TW(12)
      WK=0.
      DO 5 I=1,11
    5 WK=WK+USF(T,TW(I))*C(I)*(T-TW(I))**3
      WK=WK+C(12)*T**2+C(13)*T+C(14)
      RETURN
      END

      FUNCTION SH(T,X)
C     SPECIFIC HEAT UNITS = LBM/(FT.SEC)  CONVERTED FROM LBM/(FT.HR)
      COMMON/ASH/CWF(14),CWG(14)
      COMMON/BSH/TW(12)
      SHF=WSH(CWF,T)
      IF(X.GT.0.0001) GOTO 5
      SH=SHF
      RETURN
    5 SHG=WSH(CWG,T)
      SH=(1.-X)*SHF+X*SHG
      RETURN
      END

      FUNCTION WSH(C,T)
C     EVALUATE FUNCTION USING COEFFICIENTS DETERMINED BY
C     SPLINE FIT INTERPOLATION
      DIMENSION C(14)
      COMMON/BSH/TW(12)
      WSH=0.
      DO 5 I=1,11
    5 WSH=WSH+USF(T,TW(I))*C(I)*(T-TW(I))**3
      WSH=WSH+C(12)*T**2+C(13)*T+C(14)
```

```
      RETURN
      END

      SUBROUTINE DATAV(N)
      DIMENSION WF(12),WG(12),T1(12),T2(12)
     2,F1(12),F2(12),G1(12),G2(12)
      COMMON/AV/CWF(14),CWG(14)
      COMMON/BV/TW(12)
      DATA T1/-20.,0.,20.,40.,60.,80.,100.,120.,140.,160.,180.,200/
      DATA F1/.866,.767,.687,.620,.564,.517,.477,.441,.409,.370
     2,.329,.273/
      DATA G1/.0249,.0265,.0279,.0291,.0301,.0311,.0324,.0339,.0359
     2,.0384,.0417,.0458/
      DATA T2/-40.,-20.,0.,20.,40.,60.,80.,100.,120.,140.,160.,180./
      DATA F2/.798,.719,.654,.599,.553,.513,.480,.449,.427,.392
     2,.344,.285/
      DATA G2/.0245,.0257,.0269,.0282,.0295,.0309,.0325,.0343,.0362
     2,.0383,.0411,.045/
      I=0
      IF(N.EQ.12) I=1
      IF(N.EQ.22) I=2
      IF(I.EQ.0) STOP
      GOTO(5,15), I
    5 DO 10 I=1,12
      TW(I)=T1(I)
      WF(I)=F1(I)
   10 WG(I)=G1(I)
      GOTO 25
   15 DO 20 I=1,12
      TW(I)=T2(I)
      WF(I)=F2(I)
   20 WG(I)=G2(I)
   25 CALL COEF(TW,WF,CWF)
      CALL COEF(TW,WG,CWG)
      RETURN
      END

      SUBROUTINE DATAK(N)
      DIMENSION WF(12),WG(12),T1(12),T2(12)
     2,F1(12),F2(12),G1(12),G2(12)
      COMMON/AK/CWF(14),CWG(14)
```

```
      COMMON/BK/TW(12)
      DATA T1/-20.,0.,20.,40.,60.,80.,100.,120.,140.,160.,180.,200/
      DATA F1/.0514,.0490,.0467,.0443,.0420,.0397,.0373,.0350,.0326
     2,.0302,.0276,.0246/
      DATA G1/.0040,.0043,.0046,.0050,.0053,.0056,.0060,.0064,.0068
     2,.0072,.0076,.0083/
      DATA T2/-40.,-20.,0.,20.,40.,60.,80.,100.,120.,140.,160.,180./
      DATA F2/.0693,.0661,.0630,.0598,.0566,.0534,.0502,.0471,.0439
     2,.0407,.0371,.0318/
      DATA G2/.0040,.0044,.0048,.0052,.0056,.0060,.0064,.0068,.0072
     2,.0077,.0084,.0105/
      I=0
      IF(N.EQ.12) I=1
      IF(N.EQ.22) I=2
      IF(I.EQ.0) STOP
      GOTO(5,15), I
    5 DO 10 I=1,12
      TW(I)=T1(I)
      WF(I)=F1(I)
   10 WG(I)=G1(I)
      GOTO 25
   15 DO 20 I=1,12
      TW(I)=T2(I)
      WF(I)=F2(I)
   20 WG(I)=G2(I)
   25 CALL COEF(TW,WF,CWF)
      CALL COEF(TW,WG,CWG)
      RETURN
      END


      SUBROUTINE DATASH(N)
      DIMENSION WF(12),WG(12),T1(12),T2(12)
     2,F1(12),F2(12),G1(12),G2(12)
      COMMON/ASH/CWF(14),CWG(14)
      COMMON/BSH/TW(12)
      DATA T1/-20.,0.,20.,40.,60.,80.,100.,120.,140.,160.,180.,200/
      DATA F1/.214,.217,.220,.224,.229,.234,.240,.251,.266,.288,.317
     2,.356/
      DATA G1/.139,.145,.150,.157,.164,.174,.185,.199,.216,.235,.290
     2,.362/
      DATA T2/-40.,-20.,0.,20.,40.,60.,80.,100.,120.,140.,160.,180./
      DATA F2/.262,.266,.271,.276,.283,.291,.300,.313,.332,.357,.390
```

```
     2,.433/
      DATA G2/.146,.152,.158,.165,.175,.187,.204,.226,.253,.288,.332
     2,.376/
C     .376 - BY LINEAR EXTRAPOLATION
      I=0
      IF(N.EQ.12) I=1
      IF(N.EQ.22) I=2
      IF(I.EQ.0) STOP
      GOTO(5,15), I
    5 DO 10 I=1,12
      TW(I)=T1(I)
      WF(I)=F1(I)
   10 WG(I)=G1(I)
      GOTO 25
   15 DO 20 I=1,12
      TW(I)=T2(I)
      WF(I)=F2(I)
   20 WG(I)=G2(I)
   25 CALL COEF(TW,WF,CWF)
      CALL COEF(TW,WG,CWG)
      RETURN
      END

      SUBROUTINE COEF(X,Y,C)
      DIMENSION A(14,15),C(14),X(12),Y(12)
C     NP=12  NE=NP+2=14  NPM=NP-1=11  NPP=NP+1=13  NY=NE+1=15
C     E=3  E*(E-1)=6  E-2=1
      DO 15 J=1,12
      DO 5 I=1,11
    5 A(J,I)=USF(X(J),X(I))*(X(J)-X(I))**3
      DO 10 I=12,13
   10 A(J,I)=X(J)**(14-I)
      A(J,14)=1.
   15 A(J,15)=Y(J)
      DO 30 J=13,14
      K=1
      IF(J.EQ.14) K=12
      DO 20 I=1,11
   20 A(J,I)=USF(X(K),X(I))*6.*(X(K)-X(I))
      DO 25 I=12,14
   25 A(J,I)=(14-I)*(13-I)
   30 A(J,15)=0.
```

```
      CALL SIMEQ(14,A,C)
      RETURN
      END


      FUNCTION USF(X,XI)
C     UNIT STEP FUNCTION
      USF=0.
      IF(X.GT.XI) USF=1.
      RETURN
      END


      SUBROUTINE SIMEQ(N,A,X)
C     SUBROUTINE TO SOLVE A SYSTEM OF N LINEAR SIMULTANEOUS EQUATIONS
C     GIVEN BY THE ARRAY A(J,I) WHERE I=J+1
C     SOLUTION IS X(J)
C     ZERO  ABS VALUES LESS THAN ZERO ARE TAKEN AS 0
C     IER  IF IER IS POSITIVE SOLUTION IS VALID
C          IF IER IS NEGATIVE SOLUTION IS INVALID
      DIMENSION A(14,15),X(14)
      ZERO=1.E-6
      NP=N+1
      NM=N-1
      DO 25 I=1,NM
      IR=I
      IP=I+1
      DO 5 J=IP,N
      IF(ABS(A(IR,I)).GE.ABS(A(J,I))) GO TO 5
      IR=J
    5 CONTINUE
      IF(IR.EQ.I) GO TO 15
      DO 10 J=1,NP
      HH=A(I,J)
      A(I,J)=A(IR,J)
      A(IR,J)=HH
   10 CONTINUE
   15 IF(ABS(A(I,I)).LE.ZERO) GO TO 30
      DO 20 J=IP,N
      HH=A(J,I)
      DO 20 K=IP,NP
      A(J,K)=A(J,K)-(A(I,K)/A(I,I))*HH
   20 CONTINUE
```

```
   25 CONTINUE
      IF(ABS(A(N,N)).GT.ZERO) GO TO 40
   30 CONTINUE
C     IER=-1
      WRITE(6,35)
   35 FORMAT(/,2X,'SUBROUTINE SIMEQ - SOLUTION INVALID',/)
      RETURN
   40 DO 55 IK=1,N
      I=N+1-IK
      X(I)=A(I,NP)
      IF(I.EQ.N) GO TO 50
      IP=I+1
      DO 45 J=IP,N
   45 X(I)=X(I)-X(J)*A(I,J)
   50 X(I)=X(I)/A(I,I)
   55 CONTINUE
C     IER=1
      RETURN
      END


C     PROPERTIES OF MOIST AIR

      SUBROUTINE PSYC(Z,I)
C     SUBSCRIPT 1 DB  2 WB  3 RH  4 SH  5 H  6 DP  7  H SENS  8 H LAT
C               9 SV
C     I  2 WB  3 RH  4  SH  5  H
      DIMENSION Z(9)
      REAL H
      PMIX=1.
      PSF=2116.224
      RA=53.352
      TAZ=459.67
      IF(I .NE. 4) GO TO 10
      IF(Z(4) .GT. 0.001) GO TO 10
      WRITE(4,5) Z(1),Z(4),Z(7)
    5 FORMAT(5X,'SH .LT. 0.001',5X,3(F10.5,5X))
      Z(4)=0.001
   10 TDB=Z(1)
      IOPT=I-1
      GO TO (50,30,15,25) IOPT
   15 SH=Z(4)
      PW=PMIX*SH/(.62198+SH)
```

```
 20 PWS=PVAP(TDB)
    SHS=.62198*PWS/(PMIX-PWS)
    IF(SH.GT.SHS) SH=SHS
    PW=PMIX*SH/(.62198+SH)
    RH=PW/PWS
    GO TO 35
 25 H=Z(5)
    SH=(H-.24*TDB)/(1061.+.444*TDB)
    GO TO 20
 30 RH=Z(3)
    PWS=PVAP(TDB)
    PW=RH*PWS
    SH=.62198*PW/(PMIX-PW)
    SHS=.62198*PWS/(PMIX-PWS)
 35 SHF=SH/SHS
    SV=RA*(TDB+TAZ)*(1.+1.6078*SH)/(PMIX*PSF)
    H=.24*TDB+SH*(1061.+.444*TDB)
    WBLO=-100.
    WBHI=212.
    ITER=0
C   TWB=TDB
    TDP=DEW(TDB,PW)
    TWB=(TDB+TDP)/2.
 40 ITER=ITER+1
    PWSWB=PVAP(TWB)
    SHSWB=.62198*PWSWB/(PMIX-PWSWB)
    SHX=((1093.-.556*TWB)*SHSWB-.24*(TDB-TWB))/(1093.+.444*TDB-TWB)
    ERR=(SHX-SH)/SH
    IF(ABS(ERR).LT.0.001) GO TO 55
    IF(ITER.GE.20) GO TO 55
    IF((SHX.LT.SH).AND.(TWB.GT.WBLO)) WBLO=TWB
    IF((SHX.GT.SH).AND.(TWB.LT.WBHI)) WBHI=TWB
    IF((WBLO.LT.-99.).OR.(WBHI.GT.211)) GO TO 45
    TWB=TWB*SH/SHX
    IF(TWB.GT.TDB) TWB=TDB
 45 TWB=(WBLO+WBHI)/2.
    GO TO 40
 50 TWB=Z(2)
    PWSWB=PVAP(TWB)
    SHSWB=.62198*PWSWB/(PMIX-PWSWB)
    SH =((1093.-.556*TWB)*SHSWB-.24*(TDB-TWB))/(1093.+.444*TDB-TWB)
    PWS=PVAP(TDB)
    SHS=.62198*PWS/(PMIX-PWS)
```

```
      SHF=SH/SHS
      RH=SHF/(1.-(1.-SHF)*PWS/PMIX)
      SV=RA*(TDB+TAZ)*(1.+1.6078*SH)/(PMIX*PSF)
      H=.24*TDB+SH*(1061.+.444*TDB)
      PW=RH*PWS
C     PW=PMIX*SH/(.62198+SH)
   55 CONTINUE
      Z(2)=TWB
      Z(3)=RH
      Z(4)=SH
      Z(5)=H
      Z(6)=DEW(TDB,PW)
      Z(7)=TDB*(.24+.444*SH)
      Z(8)=SH*1061.
      Z(9)=SV
      RETURN
      END

      FUNCTION PVAP(TDB)
      TAZ=459.67
      TFREZ=491.688
      TA=TDB+TAZ
      TR=TFREZ/TA
      RTR=1./TR
      IF(TDB.LT.32.) GO TO 5
      ALOGP=10.79586*(1.-TR) + 5.02808*ALOG10(TR) + 1.50474E-4*
     2(1.-10.**(8.29692*(1.-RTR))) + 0.42873E-3*(10.**(4.76955*(1.-TR)
     3-1.) - 2.219583
      GO TO 10
    5 ALOGP=-9.096936*(TR-1.) - 3.56654*ALOG10(TR) + .876817*(1.-RTR)
     2- 2.2195983
   10 PVAP=10.**ALOGP
      RETURN
      END

      FUNCTION DEW(TDB,PW)
      PMERC=29.921
      ALFA=ALOG(PW*PMERC)
      IF(TDB.LT.32.) GO TO 5
      DEW=79.047+30.5790*ALFA+1.8893*ALFA**2
      GO TO 10
```

```
    5 DEW=71.98+24.873*ALFA+.8927*ALFA**2
      IF(DEW.GT.TDB) DEW=TDB
   10 RETURN
      END
```